Thesis/
Reports
Lin, Y. F.

INTERMOUNTAIN STATION LIBRARY
SEAM Collection

533

THE IMPLICATIONS OF COAL DEVELOPMENT ON THE ATMOSPHERIC ENVIRONMENT AND PLANT ECOSYSTEMS OF SELECTED SITES IN WESTERN NORTH DAKOTA

DOCUMENT 2C
A THERMAL POLLUTION STUDY FOR FOSSIL-FUEL POWER PLANTS

to:
SURFACE ENVIRONMENT AND MINING
U.S. FOREST SERVICE, U.S.D.A.
COOPERATIVE AGREEMENT NO. 16-418-GR

THE DEPARTMENTS OF
BOTANY, MECHANICAL ENGINEERING, SOILS

AGRICULTURAL EXPERIMENT STATION
NORTH DAKOTA STATE UNIVERSITY
FARGO, NORTH DAKOTA
NOVEMBER, 1978

A THERMAL POLLUTION STUDY FOR FOSSIL-FUEL POWER PLANTS

A Report to:

Surface, Environment and Mining (SEAM)

U.S. Forest Service USDA

(Cooperative Agreement No. 16-418-GR)

Prepared by:

Y. F. Lin, Graduate Assistant

and

Dr. Kam Li, Professor

Mechanical Engineering Department

North Dakota State University

Fargo, North Dakota 58102

Table of Contents

	PAGE
List of Tables	i.
List of Figures	iii
1. Introduction	1
2. Waste Heat Management Methods	2
3. Evaporative Cooling Towers	16
3-1 Fundamentals of evaporative cooling towers	16
3-2 Cooling system design	21
3-3 Drift	23
3-4 Evaporation	38
3-5 Formation of cooling tower fog	41
3-6 Cooling tower plumes	46
4. Cooling Pond	64
4-1 Fundamentals of cooling pond	64
4-2 Parameters affecting the pond size	70
4-3 Possible environmental impacts	78
4-4 A case study	80
5. Dry-Type Cooling Towers	92
5-1 Dry-type cooling tower fundamentals	92
5-2 Waste heat emission from a 1000 mw plant	105
5-3 Plume behavior and characteristics	107
5-4 Potential environmental impacts	120
5-5 Noise levels	130
6. References	154

LIST OF TABLES

TABLE		PAGE
3- 1	Typical computer program output	24
3- 2	Typical computer program endpoint (cont.)	25
3- 3	Typical computer program output (cont.)	26
3- 4	Typical computer program output (cont.)	28
3- 5	Typical computer program output (cont.)	29
3- 6	Size and mass distribution of dirft particles (HI-V drift eliminators)	32
3- 7	Effects of ambient air temperature on the cooling tower evaporation rate	40
3- 8	Distribution of saturation Deficit-Minneapolis (based on 10 years of data)	45
3- 9	Joint probability - stability and wind speed (%)	50
3-10	Conditional wind direction probabilities (%)	51
3-11	Effective source heights (meters) for use in dispersion calculations	52
3-12	Cooling tower performance	54
3-13	Cooling tower output conditions for various ambient temperature ranges (source strength for diffusion model)	55
3-14	Parameters of initial source description for cooling towers	57
3-15	Estimated fog frequency (hours per year)	60

TABLE		PAGE
4- 1	Condenser inlet & outlet temperature	83
4- 2	Effects of pond size on the condenser inlet outlet temperature	84
4- 3	Comparison of one-pass and two-pass condenser arrangement	85
4- 4	Annual comparative costs for various maximum pressure	86
4- 5	Cost comparisons between the one-pass and two-pass condenser arrangements	87
4- 6	Effects of pond size on the annual comparative costs	88
4- 7	Predicted values of the condenser-pond arrangement "A"	91
5- 1	Generating plants with dry-type cooling towers	93
5- 2	A typical 1000 MW fossil-fuel power plant performance	106
5- 3	Comparisons of mechanical-draft and natural-draft dry-type cooling towers for a 1000 MW plant	108
5- 4	Estimated SPC of 60 ft fans for mechanical-draft dry cooling towers	136
5- 5	Permissible noise exposures set by the Walsh-Healey Act.	140
5- 6	Acceptability criteria for noise rating number	145

LIST OF FIGURES

FIGURE		PAGE
2- 1	Heat rejected per unit of output versus the plant thermal efficiency	4
2- 2	A typical condenser-cooling-system arrangement	8
2- 3	Temperature variation of cooling tower	10
2- 4	Estimation of cooling water requirement	11
2- 5	Water consumption versus wet bulb temperature	13
2- 6	A block diagram of interrelationship of power plant environment and consumers	14
3- 1	Forced draft vertical cooling tower constructed of redwood and corrugated asbestos	18
3- 2	Effects of approach temperature on the relative size of a series of cooling towers	20
3- 3	Effects of temperature range on the relative size of a series of cooling tower	20
3- 4	Effects of ambient air wet bulb temperature on the relative size of a series of cooling towers	20
3- 5	Percent by number versus droplet size	33
3- 6	Fall velocity of water drops as function of size	34
3- 7	Drift dynamic behavior	37
3- 8	Saturation absolute humidity versus temperature	43
3- 9	Coordinate system showing Gaussian distribution in the horizontal and vertical	48

FIGURE		PAGE
3-10	Vertical dispersion coefficient as a function of downwind distance from the source	58
3-11	Horizontal dispersion coefficient as a function of downwind distance from the source	59
3-12	Contours of predicted fog frequency within 2 miles of cooling towers (20-cell towers in two separated groups)	61
3-13	Smoothed pattern of predicted fog frequency around cooling towers (20-cell towers in two separated groups)	62
4- 1	Equivalent equipment costs versus lake surface area	72
4- 2	Capability charge versus lake surface area	73
4- 3	Lake cost versus lake surface area	74
4- 4	Fuel cost versus lake surface area	75
4- 5	Fixed charge rate versus lake surface area	76
4- 6	Load factor versus surface area	77
5- 1	Three types of dry cooling systems	94
5- 2	Dry tower performance curves	100
5- 3	Estimated turbine-generator full-load heat rates	101
5- 4	Steam turbine performance curves	103
5- 5	Turbine and cooling tower combined performance curves	104
5- 6	Plume rise above a natural-draft dry cooling tower (stable conditions)	110
5- 7	Plume rise above a natural-draft dry cooling tower (neutral conditions)	112

FIGURE		PAGE
5- 8	Plume rise above a bisc 12-fan unit of mechanical-draft dry cooling tower	114
5- 9	Effects of equivalent source area on the plume rise	115
5-10	Height of Penetrable Inversion above top of natural-draft cooling tower	117
5-11	Height of Penetrable Inversion above top of mechanial-draft cooling tower	118
5-12	Entrainment ratio for a natural-draft cooling tower	119
5-13	Centerline temperature excess over ambient temperature of a heated plume from a mechanical-draft tower	121
5-14	Initial rise path of a plume from a mechanical-draft tower	122
5-15	Extent of plume normal to the ground containing a temperature excess of 1°F above the ambient	123
5-16	Extent of plume horizontal to the ground containing a temperature excess of 1°F above the ambient	124
5-17	Induced wind near a large natural-draft dry cooling tower	126
5-18	Induced wind near a large mechanical-draft dry cooling tower	127
5-19	SPL readings near an induced mechanical-draft tower	133
5-20	SPL readings near a forced mechanical-draft tower	134
5-21	Sound pressure levels for a large mechanical-draft tower at various distance from the tower compared to standard	137

FIGURE		PAGE
	noise conditions	
5-22	Noise spectrum of dry cooling towers	138
5-23	Equivalent Sound Level Contours specified in the Walsh-Healey Act	141
5-24	Noise Rating Curves	144
5-25	The measuring positions over a hemisphere surrounding the noise source	146
5-26	Diagram of one typical channel for second measurement	149

1. INTRODUCTION

The use of electrical energy in the United States has been doubling every ten years for the past several decades. To meet this growing use of electricity, many new power generating plants must be planned, designed, constructed, and operated in the near future.

The second law of thermodymanics indicates that an operation of fossil-fuel power plants will definitely result in waste effluents to the atmosphere. The magnitude of waste heat is significant. For every kw-hr of electricity generated there is approximately two kw-hr of waste heat emitted to the atmosphere. This report is intended to provide the first-order estimates of waste heat form power plants and their impacts on the environment. This study includes a survey of waste heat management methods. For each management method, information will be provided on the waste heat emission and the impacts on the environment. Whenever possible, design parameters affecting the environmental protection and the generation cost will be identified and quantitatively presented.

This report will constitute as one of the documents which are needed to directly accomplish the over-all objective No. 2 in the original proposal. The objective is

> simulate the probably dispersion of waste effluents to the atmosphere from various operation levels, and techniques of coal development at selected sites in North Dakota.

2. WASTE HEAT MANAGEMENT METHODS

The section will present the general information on the waste heat management methods. These includes the amount of waste heat generated, and methods of disposal available. In addition, the general relationships between the cooling system and power generating facility are presented. To have a good understanding of the thermal pollution controls, several frequently used technical terms are also defined in this section.

The performance of a fossil-fuel power plant can be expressed in terms of the plant net heat rate, where

$$\text{Plant Net Heat Rate} = \frac{\text{Heat Input}}{\text{Net KW Output}} \tag{2-1}$$

expressed in Btu/kWhr. By definition, the net thermal efficiency of the power plant is

$$\eta_t = \frac{\text{Net Output}}{\text{Heat Input}} \tag{2-2}$$

and related to the plant net heat rate by the equation

$$\eta_t = \frac{3413}{\text{Plant Net Heat Rate}} \tag{2-3}$$

Rearranging Equation (2-2) would give the expression for waste heat per unit of net output. That is

$$\frac{\text{Waste Heat}}{\text{Net Output}} = \frac{1}{\eta_t} - 1 \qquad (2\text{-}4)$$

Eq. (2-4) is graphically presented in Fig. (2-1). The thermal efficiency of a modern fossil-fuel power plant is largely dependent upon the steam conditions, the amount and number of reheat processes, the number of feedwater heaters and the pressure level in the condenser. In addition, there are many other factors which would also affect the optimal plant thermal efficiency. It is not the scope of the report to present the detailed calculational procedures for the plant net heat rate or thermal efficiency. Those interested are invited to contact the writer for the computer program of this purpose. [39]

It should be pointed out that not all waste heat will be rejected through the cooling system. About 10% is in fact passed through the boiler stack. Those waste heat from the cooling system is either in form of hot-dry air or in form of hot-humid air. More discussion will be in next several sections.

The magnitude of the thermal discharge is staggering. For a 1000 MW fossil-fuel power plant, the waste heat will be approximately equal to 5120 X 10^6 Btu/hr, when the plant has thermal efficiency 40% or net heat rate around 8533 Btu/Kwhr. It is obvious that concerns of the thermal discharge from power plants are certainly justified.

There are many methods available to deal with thermal discharges. One of fundamental approaches is to improve the plant thermal efficiency

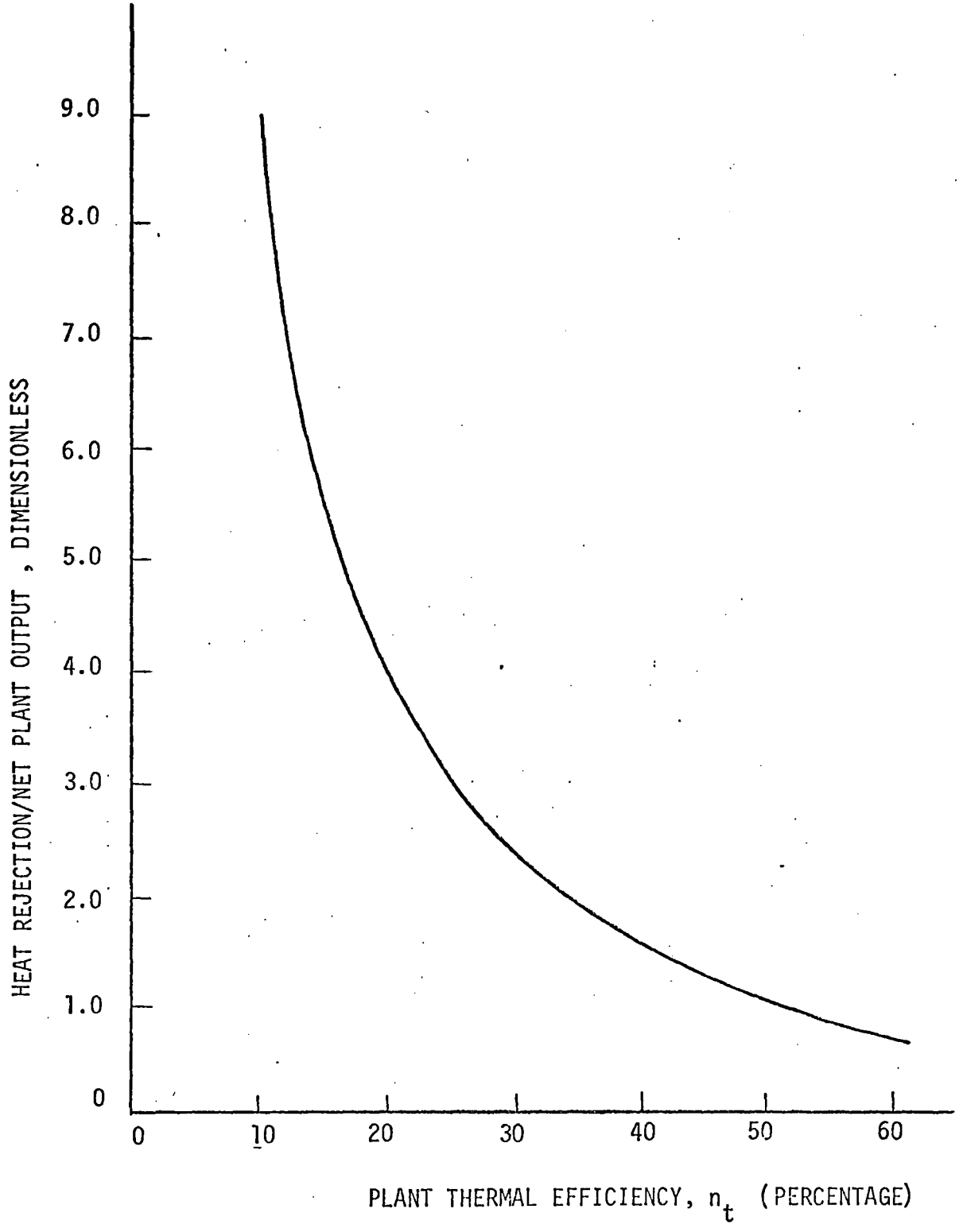


Figure 2-1. Heat rejected per unit of output versus the plant thermal efficiency

(or plant net heat rate), Changes in the plant efficiency have a marked effect on the ratio of heat rejection to power generation. As it is seen in Fig. (2-1), the ratio of waste heat generated to the net power output decreases from 2.33 to 1.0 when the efficiency improves from 30 to 50 percent. However, it does not mean that the plant efficiency should be increased indefinitely. Considerations must be given to the fact that the plant efficiency has its highest value limited by the second law of thermodynamics. The selection of plant efficiency is also affected by the economic factors under which the plant is designed,

Another approach to the reduction of thermal discharges is to utilize the waste heat. In theory, the idea is simple and attractive. In practice, there are many technical and non-technical problems to be solved. One of technical problems is the low temperature at which the waste heat is available. The matching between the waste heat supply and demand is another technical problem. The non-technical problems are obviously in the area of development planning and cooperation among the institutions. Because of these reasons, utilization of waste heat has been carried out only in a small scale in the nation,

The third fundamental approach to the thermal discharge is to dispose the waste heat. While the approach does not eliminate the waste heat, it would disperse the waste heat in such a manner that the impacts on the environment become not intolerable. Ever since 1882 when the Pearl Street Station was operated in New York, the power industry has developed a variety of cooling systems for dissipating waste heat from

thermal power plants. These cooling systems are once-through condenser cooling, evaporative cooling towers (ie,. wet towers), cooling ponds and dry-type cooling towers.

In a once-through cooling system the waste heat from power plants is transferred directly to the cooling water which would raise the temperature in the receiving water body. In some cases, this temperature rise may adversely affect the ecology of the water body. In addition, the once-through cooling system continuously requires a significant amount of water supply during operation . Because of this dependency on water availability, the plant sites are generally limited to the locations near natural water bodies such as large rivers, lakes and estuaries.

Evaporative cooling towers provide a new freedom in locating power plants. In the evaporative cooling towers the water is cooled by contacting with air. The water and air flows are directed in such a way as to provide maximum heat transfer. The heat is transferred to the air primarily by evaporation (about 75%) while the remainder is accomplished by sensible heating. The cooling water requirement for evaporative cooling towers is much smaller than that for the once-through cooling. Because of this reason, power plants with evaporative cooling towers can be located at a distance from the water body.

For economic reasons the cooling pond may be used as a cooling system for power plants. The heat transfer mechanisms in the cooling pond are similar to those of cooling towers. That is, the water is cooled by contacting with air. The lowest water temperature achievable by this method is equal to the ambient wet bulb temperature. The cooling water requirement generally referred to as the make-up water is almost the

same as that for the cooling towers. However, the land needed for pond construction is significant. In general, it takes 1 to 4 acres for one megawatt of generating facility.

As an alternative method for removing waste heat from power plants, the dry-type cooling towers have recently received some attentions. In dry cooling towers waste heat is rejected directly to the atmosphere through a system of heat exchangers. Within these heat exchangers the circulating water flows inside the finned tubes while it transfers the heat to the moving air outside by conduction and convection. In contrast to the wet cooling towers, there is no evaporation in the dry cooling towers. Because of these features, the thermal discharge to receiving waters, the evaporation and the vapor plumes of the wet tower operation would not exist in the dry cooling system. It should be pointed out that the minimum water temperature attainable by dry cooling is higher than that by the wet cooling and is equal to the ambient dry-bulb temperature. Since the heat transfer mechanism is not so effective as compared with the evaporative heat transfer, the condenser pressure will generally result in a higher level and therefore, the plant performance will be adversely affected.

To have a good understanding of cooling system, one should not isolate the cooling system from other components in the power generating system. Fig. (2-2) shows a typical arrangement. The cooling water first receives the heat in condensers and then transfers heat away in the cooling system by various heat transfer mechanisms. Except the once-through cooling, the cooling water generally goes through a closed circuit. There are many interactions among the components. In a power plant design, engineers do not look for the best performance of one compomemt, but look for the

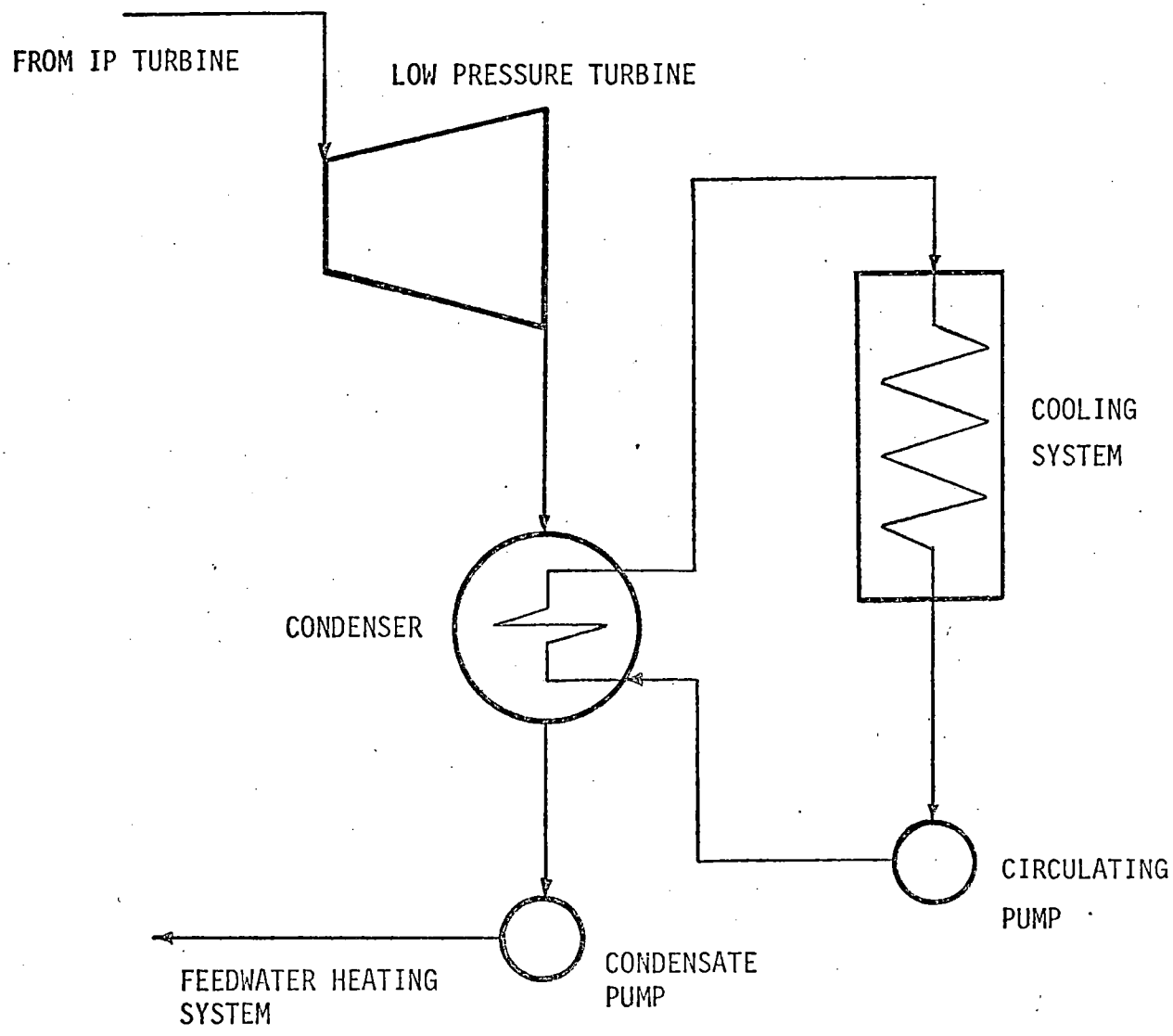


Figure 2-2. A typical condenser-cooling system arrangement

optimal performance of the overall system. Obviously, it is not within the scope of the project to prepare the engineering optimization programs. However, those interested are invited to contact the writer for the computer programs which have been prepared for other sponsored projects. [37]

Fig. (2-3) shows temperature variation in a typical condenser-cooling tower arrangement. It is seen that the cooling range is the temperature drop the water would experience when it passes through the tower. The cold water temperature is generally higher than the ambient air temperature by several degrees. The difference is defined as the cooling tower approach. For the evaporative cooling, the ambient air temperature is the wet-bulb temperature. In the dry-type cooling, the dry-bulb temperature should be used. Fig. (2-3) also shows that the cooling range is numerically equal to the temperature rise in condenser. To maintain a reasonable heat transfer surface area in the condenser, the terminal temperature difference as defined in Fig. (2-3) should be at least equal to 5°F or above.

It is of interest to know the amount of cooling water required. The estimation charts shown in Fig. (2-4) has been prepared with the following equations.

$$\dot{Q}_{waste} = 500 \dot{m} C_p DT \qquad (2\text{-}5)$$

and

$$\dot{Q}_{waste} = (PNHR - 3413) \times C_f \times (PSIZE) \qquad (2\text{-}6)$$

where $\dot{Q}_{waste}$ = waste heat from cooling system, Btu/hr.

$\dot{m}$ = cooling water flow rate, gallons per minute

DT = temperature rise in condenser, °F.

C_p = Specific heat of cooling water, Btu/lb/F.

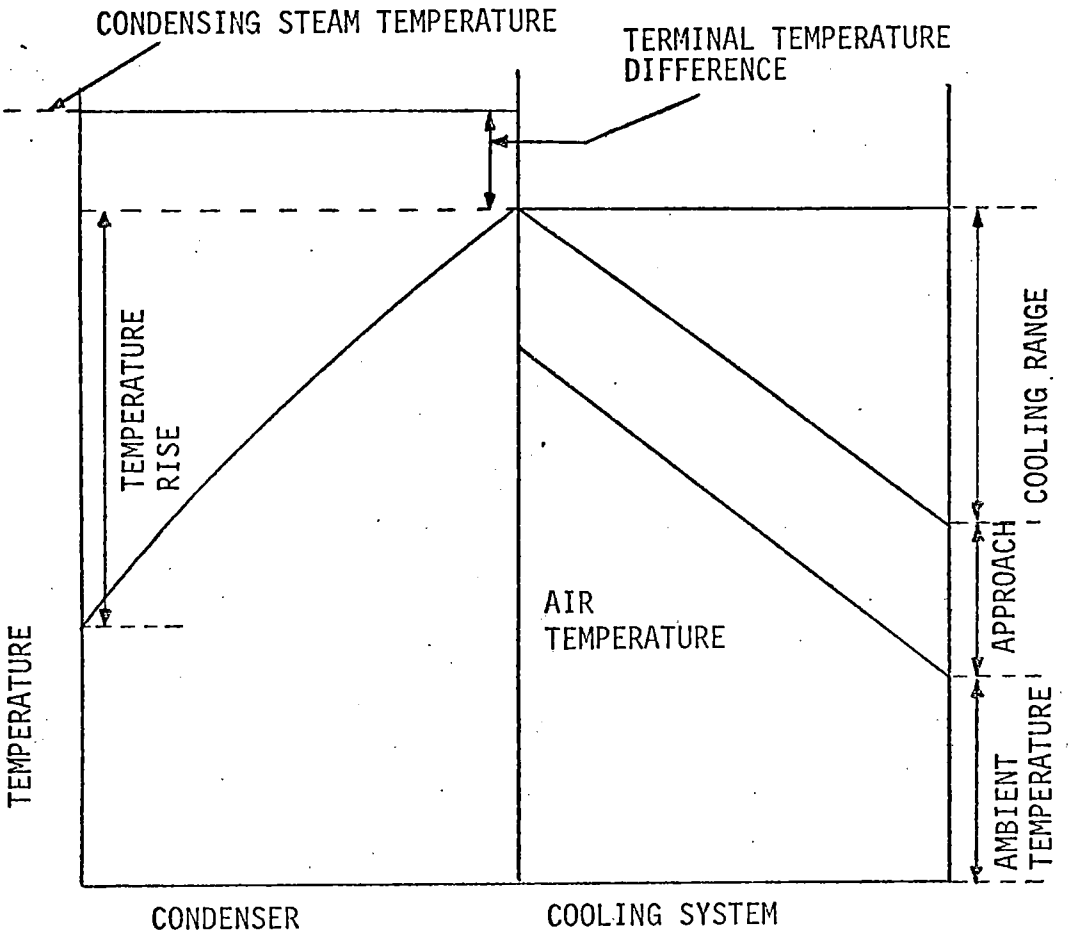


Figure 2-3. Temperature variation of cooling water

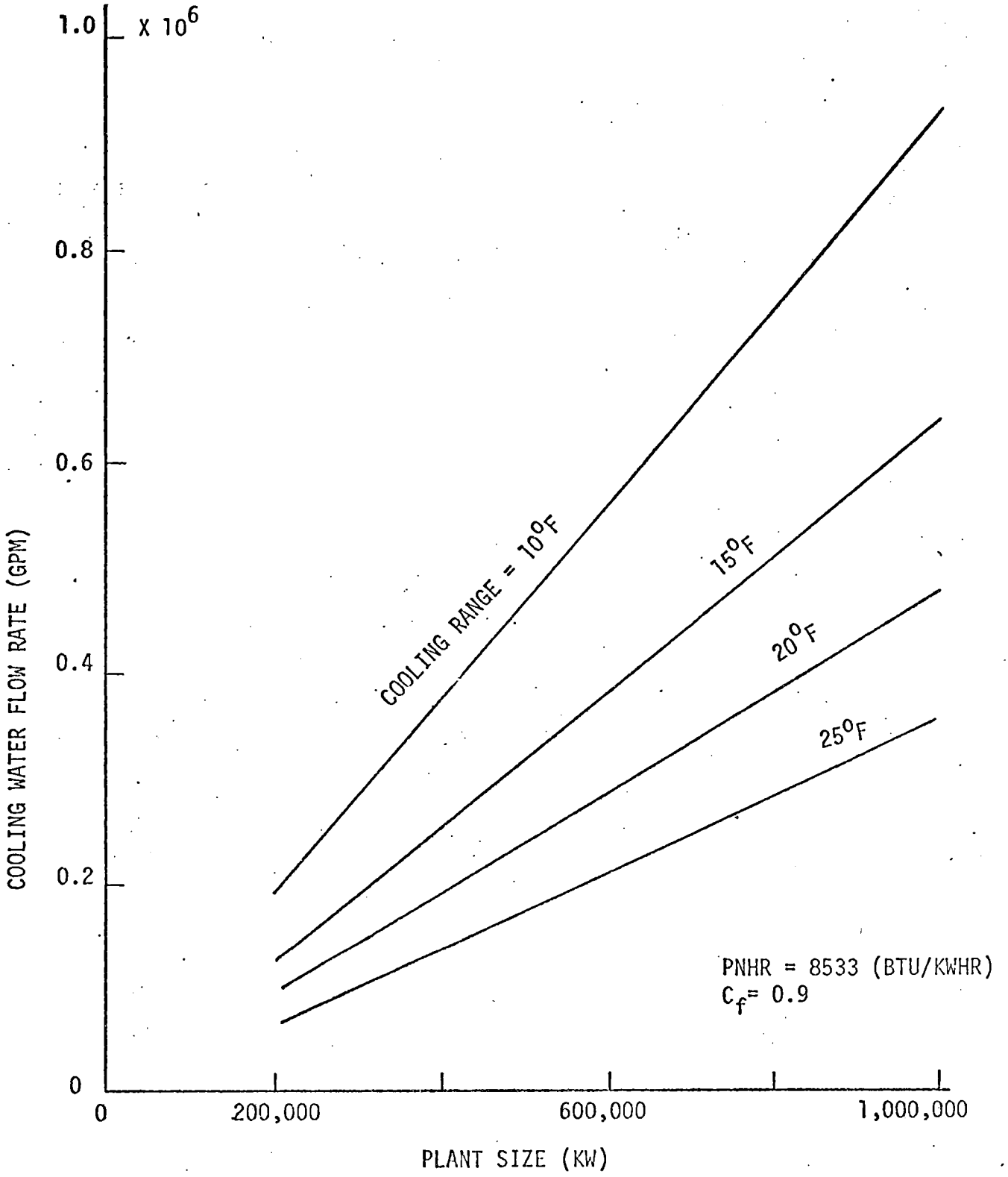


Figure 2-4. Estimation of cooling water requirement

PNHR = Plant net heat rate, Btu/Kwhr.

C_f = correction factor, dimensionless

PSIZE = Plant size, Kw.

For example, the required cooling water flow will amount to 368,000 GPM. for a 1000 mw power plant when the temperature range is taken at 25°F. It should be pointed out that this cooling water is not consumed in the process. There will be some minor losses in the system such as evaporation, drift and blowdown. But these loses would constitute only about four to five percent of the total cooling water flow. Fig. (2-5) indicates the water consumption for various cooling systems. These curves were prepared by assuming a relative humidity of 60 percent, cloud cover of 70 percent, wind speed at 8 mph. and a 20°F range with wet-bulb temperature varying from 40 to 80°F.

Fig. (2-6) shows a block diagram of interrelationships of the power plant (cooling system), environment, and consumers. It is seen that the power plant operation does not only have impacts on the quality of environment, but also affect the welfare of the consumers through the cost of electric power. These relationships must be taken into account at the time when the environment issues are discussed.

It is of interest to note in Fig. (2-6) that the operation of cooling system could be affected by the environment (through different weather). The selection of power plant cooling system power plant is a complicated matter. Climatic conditions at the plant site are important and should be taken into account. Considerations must be also given to other factors under which the plant is designed.

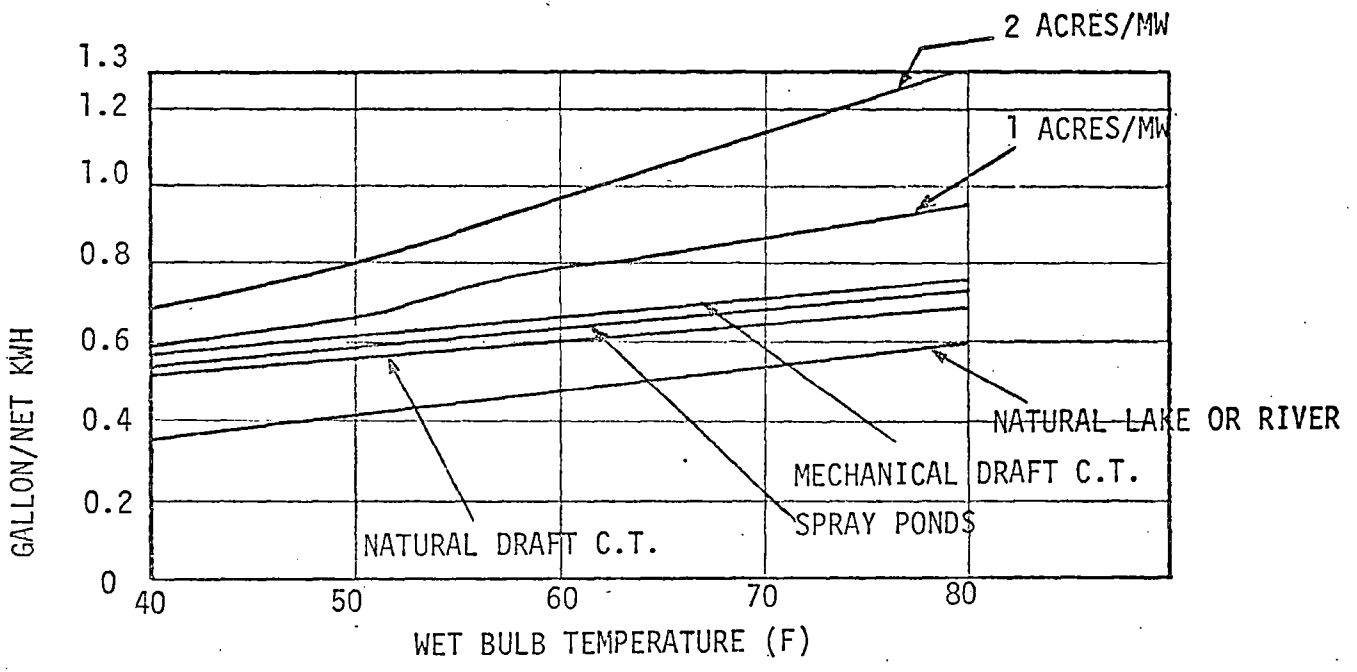


Figure 2-5. Water consumption versus wet bulb temperature

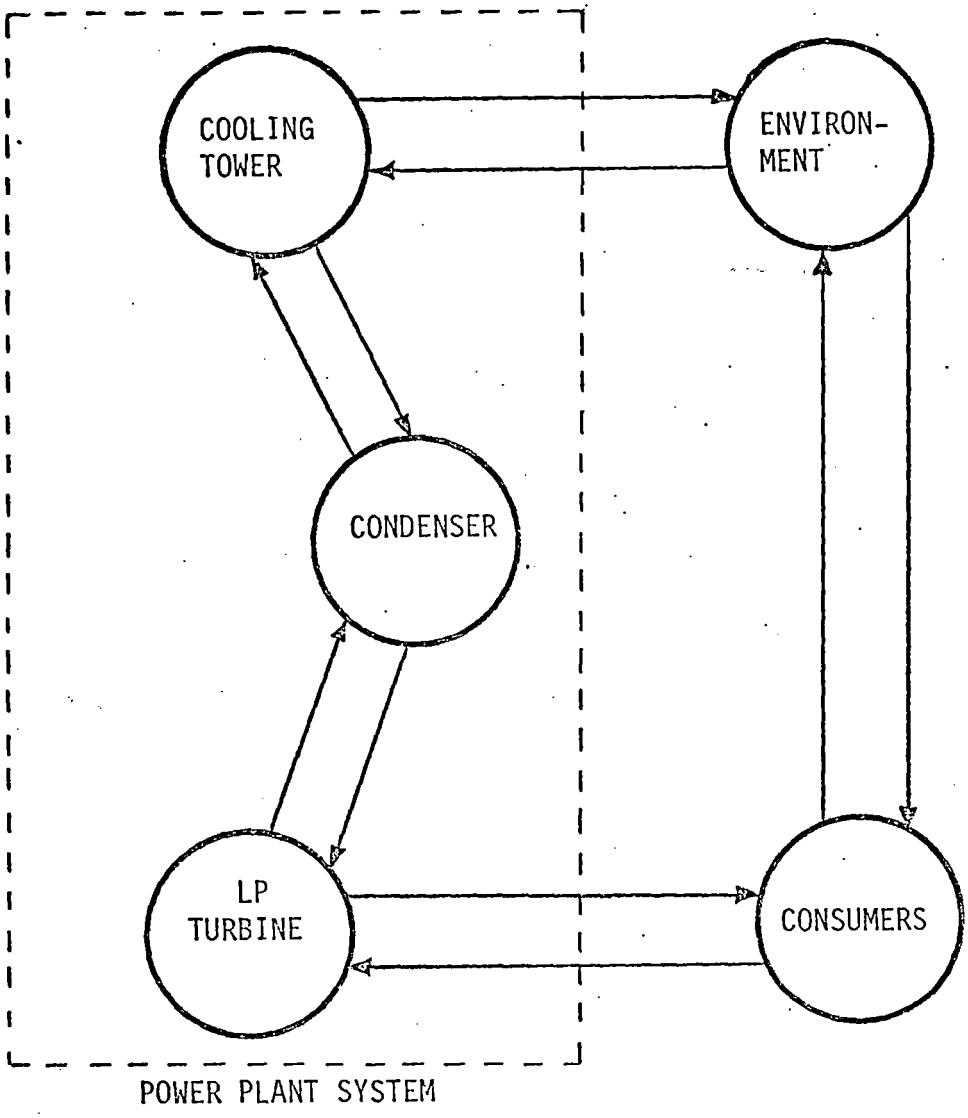


Figure 2-6. A block diagram of interrelationship of power plant, environment and consumers

In next several sections attentions will be focused on the possible impacts on environment due to the operation of power plant cooling systems. The cooling systems to be covered in this report are evaporative cooling, dry-type cooling and cooling pond.

3. EVAPORATIVE COOLING TOWERS

Cooling towers are merely heat exchangers that transfer heat from water to air. Dry-type cooling towers perform this function without direct water-air contact and rely solely upon convective heat transfer. Evaporative cooling towers are devices that cool water by bringing the water into contact with air. The water and air flows are directed in such a way as to provide maximum heat transfer. The heat is transferred to the air primarily by evaporation (about 75%) while the remainder is accomplished by sensible heating.

This section is intended to cover the fundamentals of evaporative cooling towers for power plant operations. The possible environmental impacts due to an evaporative cooling tower operation are presented. Emphasis will be placed upon the relationships between cooling system design parameters and its environmental impacts. To have a better understanding of tower operations and its impacts on environments, economic factors normally considered in design will be identified and discussed.

3-1. Fundamentals of evaporative cooling towers.

In theory, a cooling tower is a heat exchanger transferring waste heat from circulating water to the atmosphere. In actuality, it is an enclosed structure providing an intimate mixing of water and air resulting in an accelerated cooling effects by the evaporative process.

There are many different evaporative towers. They may be classified on the basis of the configuration of the fluid flow paths, The two most

common types are counter flow and cross flow. Evaporative cooling towers can also be classified according to the air-moving equipments. When the air is moved by fans, the towers are referred to as mechanical-draft cooling towers. When the air movement is induced by buoyant force, the towers will be called as natural-draft cooling towers. A cooling tower is usually comprised of the following parts:

1. Inlet-water distributor
2. Packing
3. Air moving equipment
4. Inlet-air louvres
5. Drift eliminators
6. Cold water storage basin

Fig. (3-1) indicates an induced mechanical-draft cooling tower. The rate of heat transfer depends upon several parameters. They are

1. the contact area between the water and air. Packing is often used to break the water up into smaller droplets which evaporate more easily.
2. the relative velocity between the air and water. Higher velocities would increase the heat transfer rate.
3. the residence time of the air in the packing section of the tower. Generally, sufficient packing height should be provided so that the air at the tower exit will be saturated.

The cooling tower theory is based upon the principles governing the mass and heat transfer between the water droplets and its surrounding air.

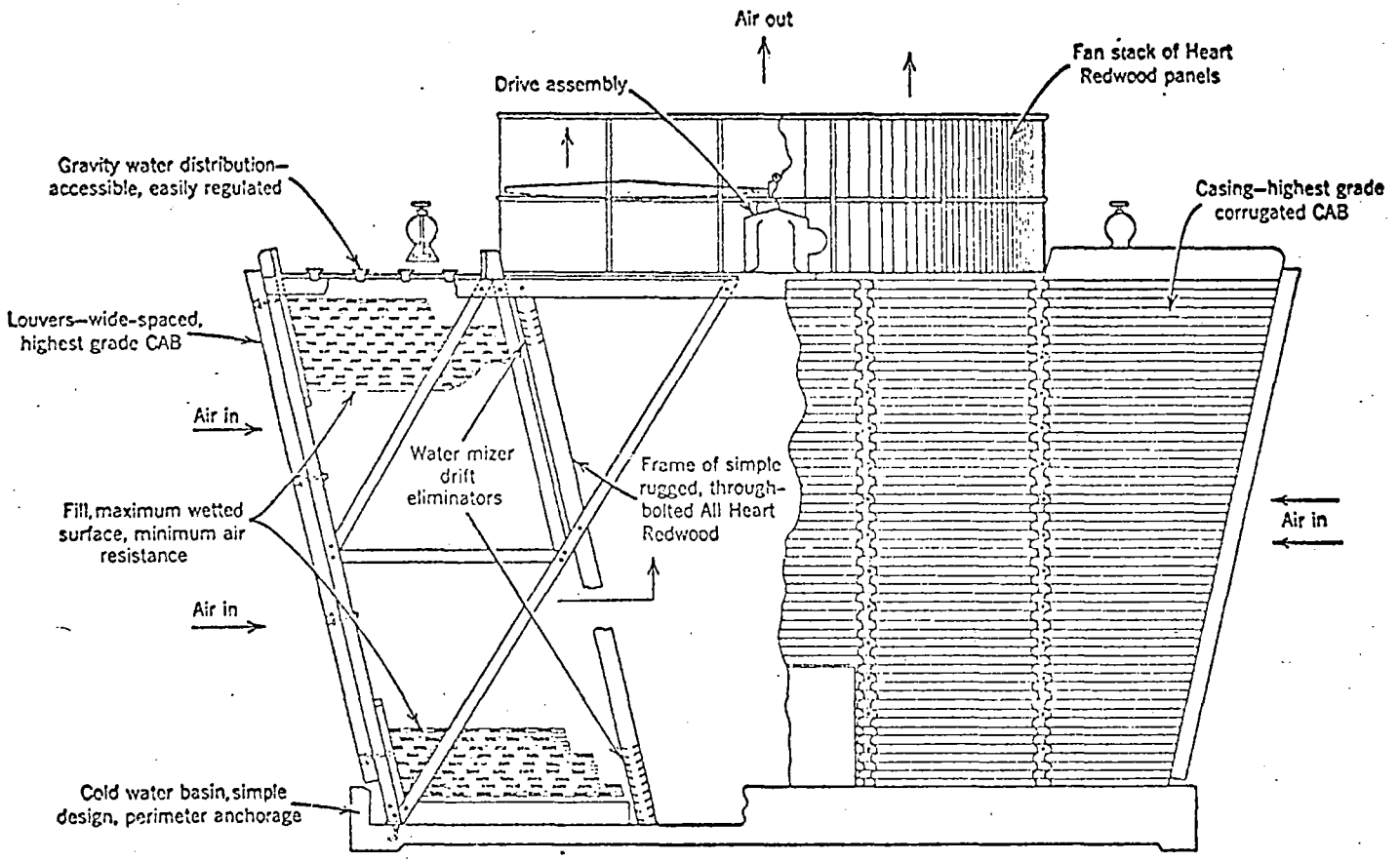

Air out
Drive assembly
Fan stack of Heart Redwood panels
Gravity water distribution—accessible, easily regulated
Casing—highest grade corrugated CAB
Louvers—wide-spaced, highest grade CAB
Air in
Water mizer drift eliminators
Frame of simple rugged, through-bolted All Heart Redwood
Fill, maximum wetted surface, minimum air resistance
Air in
Air in
Cold water basin, simple design, perimeter anchorage

Figure 3-1. Forced draft vertical cooling tower constructed of redwood and corrugated asbestos

The equation developed by Merkel in 1925 is still the most fundamental equation used to describe the cooling tower performance. For reference, it is expressed below [46]

$$\frac{KaV}{L_w} = \int_{twl}^{tw2} \frac{dtw}{H_s - H_a} \qquad (3\text{-}1)$$

where K = enthalpy transfer coefficient

a = contact surface area per unit vol.

V = packing volume

twl = cold water temperature

tw2 = hot water temperature

tw = water temperature

H_s = enthalpy of saturated air at the load water temperature

H_a = enthalpy of the air stream

L_w = water flow rate

An analysis of Eq. (3-1) would reveal that the tower size is a function of the following four variables: (1) cooling range, (2) tower approach, (3) ambient wet-bulb temperature, (4) heat load. All these terms have been defined previously in this report. To show the general trend of these effects, calculations have been made for cooling tower size relative to that for a reference unit evaluated for 75°F wet-bulb, 25F cooling range and a 10F tower approach. The results are shown in Fig. (3-2), Fig. (3-3), and Fig. (3-4). The tower size is linearly proportional to the heat load. Because of this, the curve is omitted from presentation.

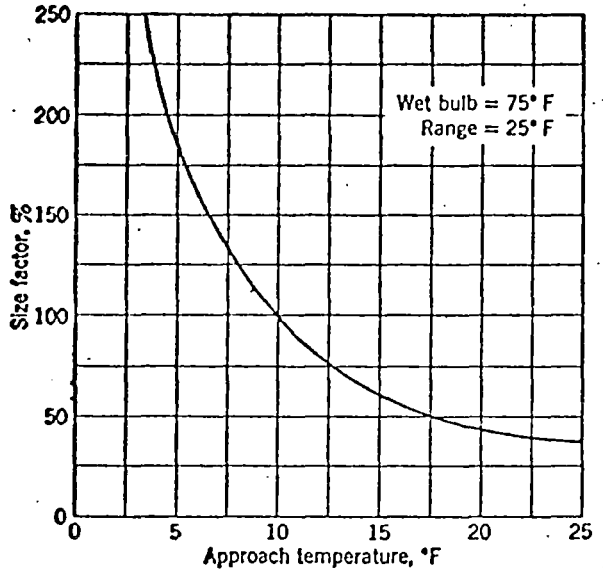


Figure 3-2. Effects of approach temperature on the relative size of a series of cooling towers.

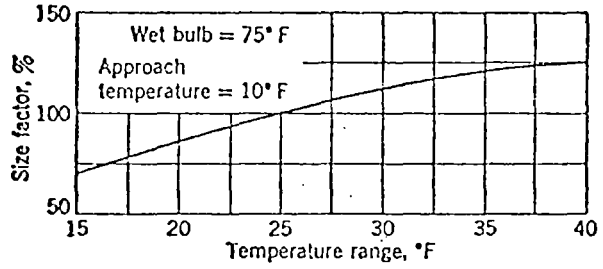


Figure 3-3. Effects of temperature range on the relative size of a series of cooling towers.

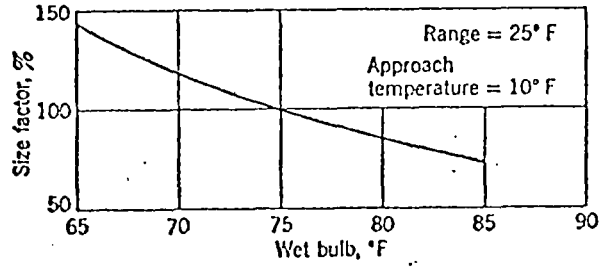


Figure 3-4. Effects of ambient air wet bulb temperature on the relative size of a series of cooling towers.

The variables affecting the tower size are also the variables which would affect the conditions of effluents from cooling towers. Discussion on this aspect will be presented later.

3-2. Cooling System Design.

The power plant cooling system consists of condensers, cooling towers, circulating pumps and other associated equipment. To have optimal performance and minimum impacts on environment, all components must be integrated into a system and system optimization procedures must be taken. In the following are delineations of the interrelationships among these components.

As seen in Fig. (3-2), the tower size would decrease as the tower approach increase. That means that the tower cost would be reduced with large tower approach. For a given cooling range, this means a higher condenser pressure and thus a higher fuel cost. If the condenser pressure remains constant, an increase of tower approach would reduce the cooling range and thus, increase the circulating water flow rate. In an over-all evaluation the condenser size will increase and more power will be required for the circulating water pumps. Therefore, the saving in the cooling tower cost must be balanced by the increase of fuel cost, and condenser cost.

Fig. (3-3) indicates that the cooling tower size would decrease with a decrease of cooling range. In addition to the savings in the tower cost, a small cooling range will result at a low condenser pressure and thus at a saving of fuel. However, associated with a small cooling range,

there is an increase of cooling water flow rate and thus an increase in condenser size and pump works required. Again, the balance between these two factors must be made in design.

The cooling towers are usually designed to operate under certain site conditions. To an evaporative cooling tower, the ambient wet-bulb temperatures is important. Fig. (3-4) indicates the effects of ambient wet-bulb temperature on the tower size required. It is seen that as the ambient wet-bulb temperature increases, the required tower size would decrease. In other words, the evaporative cooling tower would perform better at a high ambient wet-bulb temperature and poorly at a low wet-bulb temperature. These, in turn, would affect the condenser pressure and thus fuel cost of power plant operation. Because of these reasons, the cooling system design must involve a system simulation and determination of over-all performance under various operation and weather conditions.

The heat load in condenser is identical to the heat load in the cooling tower. The heat load generally determined by the plant size would have a direct impact on initial and operating costs. However, the relation between them is not linear. As the heat load increases, the unit cost of cooling system usually would decrease. These does not mean that the plant size can be increased indefinitely. As indicated later, the environmental impact factors would impose a limitation on the plant size.

The cooling system design is a complicated process. The selection of cooling system parameters such as cooling range, and tower approach requires an optimization calculation usually done on the digital computer. Computer programs have been prepared for these purposes. In these programs

the cooling systems are evaluated in the following major areas:

1. initial investments
2. operating cost
3. plant fuel cost
4. generating capability penality
5. capitalized annual cost

The final results are presented in terms of annual owning and operating costs. For reference, typical computer outputs are included and presented in Table 3-1 to table 3-5. The document of computer programs was presented in reference [37].

3-3. Drift

Cooling tower drift has been traditionally defined as mechanically entrained water droplets which are generated inside the cooling tower and carried along with the air flowing through the tower and exhausted to the environment. Based upon this definition, one would expect these water particles to have essentially the same chemical composition as the circulating water in the cooling tower. Most drift loss guarantees are quoted as a percent of the circulating water rate. It ranges from 0.01 ~ 0.05%. [58]

In the evaluation of drift and its potential environmental hazard, we are ultimately interested in the total quantity of drift droplets discharged to the environment, their chemical impurities, and the subsequent behavior of this drift as it interacts with the environment. This section deals exclusively with the mechanically entrained water

Table 3-1 Typical output of Computer Program No. 1

COLD END OPTIMIZATION PROGRAM
OUTPUT: TABLE 1
SUMMARY OF COMPARATIVE STUDY

MAXIMUM BACK PRESSURE @V#0 AND 5% O.P.@ 78.0 DEG. F W.B.	4.5				
SUMMARY OF PHYSICAL DATA:					
COOLING TOWER APPROACH TO W.B.	10.00	12.00	14.00	16.00	18.00
COOLING TOWER RANGE - DEG F	33.75	32.20	30.64	29.07	27.49
CONDENSER SURFACE- SQ FT	313979.56	329043.81	345837.75	364516.38	385414.88
MAXIMUM GENERATING CAPABILITY- MW GROSS	882.57	882.57	882.57	882.57	882.57
AUXILIARY POWER CONSUMPTION- MW	9.95	9.68	9.57	9.61	9.71
THE NET MAXIMUM GENERATING CAPABILITY- MW	872.62	872.89	873.00	872.95	872.86
THE DIFFERENTIAL NET MAXIMUM GENERATING CAPABILITY-MW(BASE: 900.)	-27.38	-27.11	-27.00	-27.05	-27.14
SUMMARY OF ANNUAL COSTS--$1000					
FIXED CHARGES OR INVESTMENT @ 0.16	8724.18	8688.39	8684.77	8713.35	8754.00
OPERATING COSTS	409.43	402.05	400.63	405.17	411.84
FUEL COST RESULTING FROM CONDENSER BACK PRESSURE	21193.59	21193.43	21193.22	21193.38	21193.47
GENERATING CAPABILITY PENALTY@ $30./KWYR	821.50	813.34	809.96	811.35	814.26
TOTAL COMPARATIVE ANNUAL COST	31148.69	31097.20	31088.57	31123.24	31173.56
CAPITALIZED ANNUAL COST @ 0.16 F.C.RATE- $1000	194679.31	194357.50	194303.56	194520.25	194834.75
DIFFERENTIAL CAPITALIZED COST- $1000	375.75	53.94	0.0	216.69	531.19

Table 3-2 Typical output of Computer Program No. 1

COLD END OPTIMIZATION PROGRAM
OUTPUT: TABLE 2
COMPARATIVE INITIAL COSTS OF COLD END SYSTEMS

ESTIMATED INVESTMENT					
CONDENSER SURFACE: SQ.FT.	313979.56	329043.81	345837.75	364516.38	385414.88
NUMBER OF TUBES: 1.000 IN. O.D.	28555.	29925.	31452.	33151.	35052.
SHELL BASE PRICE:	256596.88	263995.06	272242.63	281416.00	291679.19
TOTAL SHELL PRICE:	513193.75	527990.13	544485.25	562832.00	583358.38
TUBE LENGTH EXTRA:	112902.25	116157.44	119786.38	123822.63	128338.44
WATER BOX PRESSURE EXTRA:	9300.19	9970.26	10616.93	11336.18	12140.90
HOT WELL VOLUME:	33670.39	34845.40	36155.32	37612.26	39242.34
EXPANSION JOINTS:	51000.00	51000.00	51000.00	51000.00	51000.00
EXTRACTION HEATER PROVISION:	22000.00	22000.00	22000.00	22000.00	22000.00
EXTENDED NECK 12 FEET:	50000.00	50000.00	50000.00	50000.00	50000.00
SUBTOTAL:	792156.56	811963.19	834043.81	858603.00	886080.00
CONDENSER PRICE (ESCALATED FACTOR@ 1.50):	1188234.00	1217944.00	1251065.00	1287904.00	1329120.00
INSTALLATION COST:	297058.50	304486.00	312766.25	321976.00	332280.00
TUBE MATERIAL	847744.75	888418.19	933761.81	984194.13	1040620.06
TUBE INSTALLATION COST	57110.06	59850.12	62904.78	66302.25	70103.50
TOTAL CONDENSER COST:	2390146.00	2470698.00	2560497.00	2660376.00	2772123.00
CIRC. WATER PUMPS	580329.19	608172.50	639212.69	673736.69	712363.19
CIRC. WATER PUMP MOTORS:	382947.00	399600.56	418121.31	438669.69	461600.50
ELECTRICAL WORK:	191473.50	199800.25	209060.63	219334.81	230800.25
CIR. WATER PIPE:	1578291.00	1644409.00	1718129.00	1800124.00	1891861.00
CONCRETE STRUCTURES:	1357321.00	1414191.00	1477590.00	1548106.00	1627000.00
COOLING TOWER:	2538645.00	2194688.00	1934654.00	1756453.00	1596637.00
COOLING TOWER FOUNDATIONS:	126932.19	109734.38	96732.69	87822.63	79831.81
SUBTOTAL:	6755928.00	6570594.00	6493498.00	6524244.00	6589592.00
CIRC. AND COOLING SYSTEM PRICE (ESCALATED FACTOR@ 1.40)	9458296.00	9198829.00	9090894.00	9133939.00	9225428.00
MAIN STEAM TURBINE COST	31772464.00	31772464.00	31772464.00	31772464.00	31772464.00
TOTAL INVESTMENT COST -$1000	43620.89	43441.98	43423.84	43566.77	43770.00
INDIRECT & OVERHEAD COSTS -$1000 @0.25	10905.22	10860.49	10855.96	10891.69	10942.50
TOTAL ESTIMATED COMPARABLE INITIAL COST: -$1000	54526.12	54302.47	54279.80	54458.46	54712.50

Table 3-3 Typical output of Computer Program No. 1

COLD END OPTIMIZATION PROGRAM
OUTPUT: TABLE 3
NET STATION HEAT RATES

	4.50				
MAXIMUM EXPECTED LOAD- VWO & 5% O.P.					
HEAT LOAD: MBTU/HR	4080.00	4080.00	4080.00	4080.00	4080.00
APPROACH TO 78.0WET BULB- DEG.F	10.00	12.00	14.00	16.00	18.00
TERMINAL TEMPERATURE DIFFERENCES	8.03	7.58	7.14	6.71	6.29
TUBE LENGTH- FEET	42.00	42.00	42.00	42.00	42.00
GALLONS PER MINUTE	241803.88	253405.25	266338.69	280723.69	296818.06
SURFACE AREA- SQ.FT.	313979.56	329043.81	345837.75	364516.38	385414.88
CIRCULATING WATER PIPE DIAMETER- FT.	7.56	7.74	7.93	8.14	8.37
GROSS GENERATING CAPABILITY- MW	882.57	882.57	882.57	882.57	882.57
STEAM TURBINE EXHAUST ANNULUS AREA- SQ FT	66.10	66.10	66.10	66.10	66.10
NUMBER OF FLOWS AT THE EXHAUST END	4.00	4.00	4.00	4.00	4.00
SUMMER OPERATION (RELATIVE HUMIDITY@ 50.%)					
POWER GENERATION- MW	800.00	800.00	800.00	800.00	800.00
WET BULB TEMP- DEG F	71.00	71.00	71.00	71.00	71.00
HEAT LOAD- MBTU/HR	3850.00	3850.00	3850.00	3850.00	3850.00
BACK PRESSURE- IN. HG. ABS.	3.60	3.60	3.60	3.60	3.59
NET STATION HEAT RATE- BTU/KWH	9700.52	9700.31	9699.95	9699.63	9699.31
POWER GENERATION- MW	600.00	600.00	600.00	600.00	600.00
WET BULB TEMP- DEG F	71.00	71.00	71.00	71.00	71.00
HEAT LOAD- MBTU/HR	3150.00	3150.00	3150.00	3150.00	3150.00
BACK PRESSURE- IN. HG. ABS.	2.80	2.80	2.80	2.80	2.79
NET STATION HEAT RATE- BTU/KWH	9726.46	9726.70	9726.31	9726.36	9725.97
POWER GENERATION- MW	400.00	400.00	400.00	400.00	400.00
WET BULB TEMP- DEG F	71.00	71.00	71.00	71.00	71.00
HEAT LOAD- MBTU/HR	2340.00	2340.00	2340.00	2340.00	2340.00
BACK PRESSURE- IN. HG. ABS.	2.07	2.09	2.10	2.12	2.14
NET STATION HEAT RATE- BTU/KWH	9727.70	9731.98	9736.10	9740.19	9744.25

Table 3-3 Typical output of Computer Program No. 1 (continued)

SPRING- FALL OPERATION (RELATIVE HUMIDITY@ 50.%)					
POWER GENERATION- MW	800.00	800.00	800.00	800.00	800.00
WET BULB TEMP- DEG F	56.00	56.00	56.00	56.00	56.00
HEAT LOAD- MBTU/HR	3850.00	3850.00	3850.00	3850.00	3850.00
BACK PRESSURE- IN. HG. ABS.	2.57	2.55	2.54	2.54	2.53
NET STATION HEAT RATE- BTU/KWH	9606.63	9605.84	9605.19	9604.93	9604.43
POWER GENERATION- MW	600.00	600.00	600.00	600.00	600.00
WET BULB TEMP- DEG F	56.00	56.00	56.00	56.00	56.00
HEAT LOAD- MBTU/HR	3150.00	3150.00	3150.00	3150.00	3150.00
BACK PRESSURE- IN. HG. ABS.	1.97	1.96	1.95	1.95	1.94
NET STATION HEAT RATE- BTU/KWH	9604.75	9604.28	9604.08	9603.79	9603.67
POWER GENERATION- MW	400.00	400.00	400.00	400.00	400.00
WET BULB TEMP- DEG F	56.00	56.00	56.00	56.00	56.00
HEAT LOAD- MBTU/HR	2340.00	2340.00	2340.00	2340.00	2340.00
BACK PRESSURE- IN. HG. ABS.	1.43	1.43	1.42	1.43	1.44
NET STATION HEAT RATE- BTU/KWH	9612.55	9612.41	9612.30	9612.63	9612.96
WINTER OPERATION (RELATIVE HUMIDITY@ 50.%)					
POWER GENERATION- MW	800.00	800.00	800.00	800.00	800.00
WET BULB TEMP- DEG F	41.00	41.00	41.00	41.00	41.00
HEAT LOAD- MBTU/HR	3850.00	3850.00	3850.00	3850.00	3850.00
BACK PRESSURE- IN. HG. ABS.	1.88	1.85	1.83	1.80	1.79
NET STATION HEAT RATE- BTU/KWH	9576.79	9576.28	9575.81	9575.41	9575.09
POWER GENERATION- MW	600.00	600.00	600.00	600.00	600.00
WET BULB TEMP- DEG F	41.00	41.00	41.00	41.00	41.00
HEAT LOAD- MBTU/HR	3150.00	3150.00	3150.00	3150.00	3150.00
BACK PRESSURE- IN. HG. ABS.	1.42	1.40	1.39	1.37	1.36
NET STATION HEAT RATE- BTU/KWH	9584.70	9584.49	9584.37	9584.19	9584.10
POWER GENERATION- MW	400.00	400.00	400.00	400.00	400.00
WET BULB TEMP- DEG F	41.00	41.00	41.00	41.00	41.00
HEAT LOAD- MBTU/HR	2340.00	2340.00	2340.00	2340.00	2340.00
BACK PRESSURE- IN. HG. ABS.	1.00	0.99	0.98	0.98	0.97
NET STATION HEAT RATE- BTU/KWH	9599.82	9599.58	9599.22	9599.03	9598.85

Table 3-4 Typical output of Computer Program No. 1

COLD END OPTIMIZATION PROGRAM
OUTPUT: TABLE 4
ANNUAL PLANT FUEL COSTS

MAXIMUM BACK PRESSURE @ VWO AND 5% OP @ 78.0 DEG. F W.B. 4.5

SUMMER OPERATION

LOAD (MW)	HOURS	FUEL COST -$1000				
800.00	600.00	2328.12	2328.07	2327.99	2327.91	2327.83
600.00	900.00	2626.14	2626.21	2626.10	2626.11	2626.01
400.00	500.00	972.77	973.20	973.61	974.02	974.42

SPRING- FALL OPERATION

LOAD (MW)	HOURS	FUEL COST -$1000				
800.00	600.00	2305.59	2305.40	2305.24	2305.18	2305.06
600.00	1400.00	4033.99	4033.79	4033.71	4033.59	4033.54
400.00	2000.00	3845.02	3844.96	3844.92	3845.05	3845.18

WINTER OPERATION

LOAD (MW)	HOURS	FUEL COST -$1000				
800.00	300.00	1149.21	1149.15	1149.10	1149.05	1149.01
600.00	700.00	2012.79	2012.74	2012.72	2012.68	2012.66
400.00	1000.00	1919.96	1919.91	1919.84	1919.80	1919.77

ANNUAL OPERATION HOUR 8000.000

ANNUAL PLANT FUEL COST (-$1000)					
	21193.59	21193.43	21193.22	21193.38	21193.47

Table 3-5 Typical output of Computer Program No. 1

COLD END OPTIMIZATION PROGRAM
OUTPUT: TABLE 5
OPERATION COSTS AND GENERATING CAPABILITY PENALTIES

MAXIMUM BACK PRESSURE @VWO AND 5% O.P.@ 78.0 DEG. F W.B.	4.5				
OPERATION COSTS					
CIRCULATING WATER PUMPS:					
COOLING TOWER HEAD- FT.	48.00	48.00	48.00	48.00	48.00
CONDENSER LOSS- FT.	26.70	26.70	26.70	26.70	26.70
TUBES	23.00	23.00	23.00	23.00	23.00
WATER BOX OUTLET	0.50	0.50	0.50	0.50	0.50
WATER BOX INLET	1.00	1.00	1.00	1.00	1.00
TUBE ENDS	2.20	2.20	2.20	2.20	2.20
PUMP SUCTION- FT.	0.50	0.50	0.50	0.50	0.50
VELOCITY HEAD- FT.	1.00	1.00	1.00	1.00	1.00
PIPE FRICTION (5000. FT)- FT.	17.75	17.34	16.92	16.48	16.02
MISC. FITTINGS- FT.	2.00	2.00	2.00	2.00	2.00
TOTAL PUMP HEAD- FT.	95.95	95.54	95.12	94.68	94.22
FLOW RATE-GPM	241803.88	253405.25	266338.69	280723.69	296818.06
PUMP POWER- HP	8062.04	8412.64	8802.55	9235.15	9717.91
PUMP OPERATING COST--$1000/YR	268.69	280.38	293.37	307.79	323.88
COOLING TOWER FANS:					
TOWER UNITS	338486.13	292625.19	257953.88	234193.88	211551.63
POWER REQUIRED PER EACH T.U.- BHP	0.0120	0.0120	0.0120	0.0120	0.0120
FAN POWER- BHP	5278.54	4563.36	4022.67	3652.15	3299.05
FAN OPERATING COST--$1000/YR	140.74	121.67	107.25	97.38	87.96
GENERATING CAPABILITY PENALTIES					
MAXIMUM GENERATING CAPABILITY- MW GROSS	882.57	882.57	882.57	882.57	882.57
AUXILIARY POWER CONSUMPTION- MW	9.95	9.68	9.57	9.61	9.71
THE NET MAXIMUM GENERATING CAPABILITY- MW	872.62	872.89	873.00	872.95	872.86
THE DIFFERENTIAL NET MAXIMUM GENERATING CAPABILITY- MW	-27.38	-27.11	-27.00	-27.05	-27.14
GENERATING CAPABILITY PENALTY--$1000/YR	821.50	813.34	809.96	811.35	814.26

droplets as drift and its environmental impact.

The cooling tower drift is small when it is expressed in term of percentage of the circulating water rate. But in modern power plant operation (800 MW) the circulating water rate frequently amounts to 300,000 GPM or above. The drift loss would be approximately equal to 150 gallons per minutes (calculated at 0.05% loss). In general, the drift loss is determined by a field measurement. The drift measurement system is designed to physically extract a continuous isokinetic sample from the cooling tower exhaust air along with the entrained drift particles. The drift particles are separated from the exhaust air and collected for quantitative measurement and analysis at the conclusion of the test. The measurement system generally consists of two major functional components, isokinetic sampler and cyclone separator. More information of drift measurement system can be found in reference [4].

To understand the dynamic and thermodynamic behavior of the drift as it interacts with the environment, one should know the drift particle size and mass distribution. Several of the measurement techniques used in cloud physics were evaluated to establish the most effective method for measuring cooling tower drift drop size. The impaction slide methods have showed the most promising [4], because they yield accurate results down to the lower size limit of natural fog. The measurement technology, calibration, and correlation have been well established by cloud physics investigators. Also, these techniques are easily adaptable to field measurement on cooling towers and a permanent record of drop size can be obtained.

Table 3-6 presents the typical size and mass distribution of drift particles. For reference, the drop size distribution is also shown in Fig. (3-5). It should be noted that these field measurements were made on towers where the total drift loss was measured as 0.001% of the circulating water rate and therefore are representative of the current state of the art of drift eliminator designs.

Examination of these results reveals several important aspects of drift. First, the exhaust drop size distribution is bimodal with peaks in the 35 micron and 300 micron size ranges. It is in contrast to the natural atmospheric aerosols that would have a unimodal size distribution. Secondly, a few drops in the 1000 to 2400 micron range are present in the exhaust air. Even a casual field observation shows that water droplets in this size range are emitted from a cooling tower since they are clearly visible and easily detected. Tests indicated that these large droplets were generally formed in the tower plenum area.

With the information of drift particle size and mass distribution, it is now possible to assess the significance of drift in term of its potential to create negative effects on the environment or plant equipment in the vicinity of the cooling tower. In general, the dynamic behavior of drfit is a complicated function of the original droplet size, condensation or evaporation rates, aerodynamic and gravitational forces and meteorological conditions. Of interest is the terminal fall velocity of a droplet which can be determined by balancing the aerodynamic drop force with the weight of the droplet. Fig. (3-6) indicates the typical curve of terminal fall velocity vs. the droplet size.

TABLE 3-6

SIZE AND MASS DISTRIBUTION OF DRIFT PARTICLES
(HI-V DRIFT ELIMINATORS)

DROPLET DIAMETER (MICRON)	% OF SAMPLE BY NUMBER	% MASS BY DROPLET SIZE
22	24.0	0.43
29	36.0	1.49
44	26.0	3.76
58	6.3	2.09
65	4.0	1.86
87	1.4	1.56
108	0.67	1.43
120	0.43	1.26
132	0.28	1.09
144	0.26	1.32
174	0.65	5.81
300	0.11	5.04
450	0.027	4.17
600	0.011	4.01
750	0.0055	4.00
900	0.0033	4.03
1050	0.0024	4.57
1200	0.0019	5.46
1350	0.0016	6.80
2250	0.00095	17.99
2400	0.0010	21.83

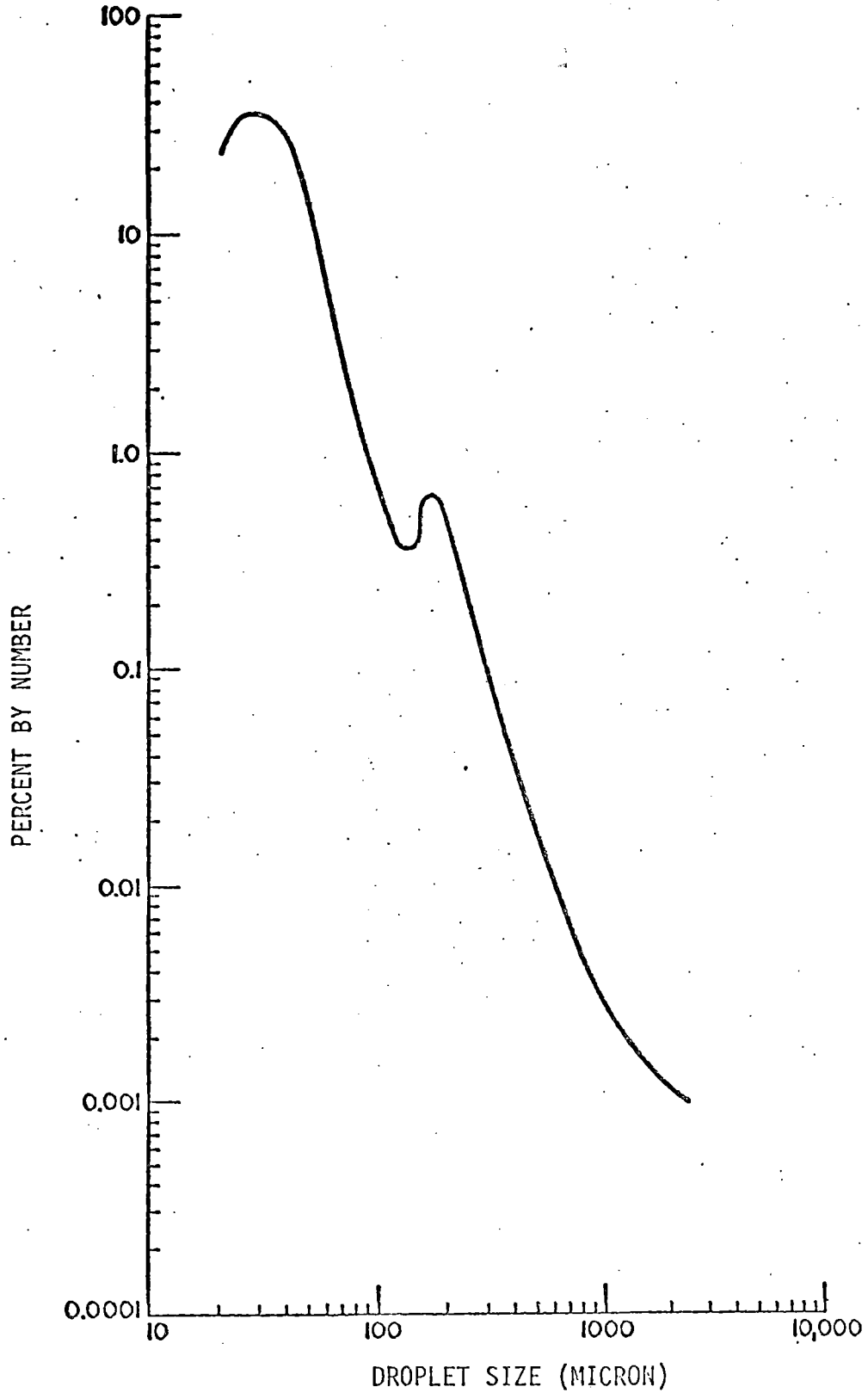


Figure 3-5. Percent by number versus droplet size

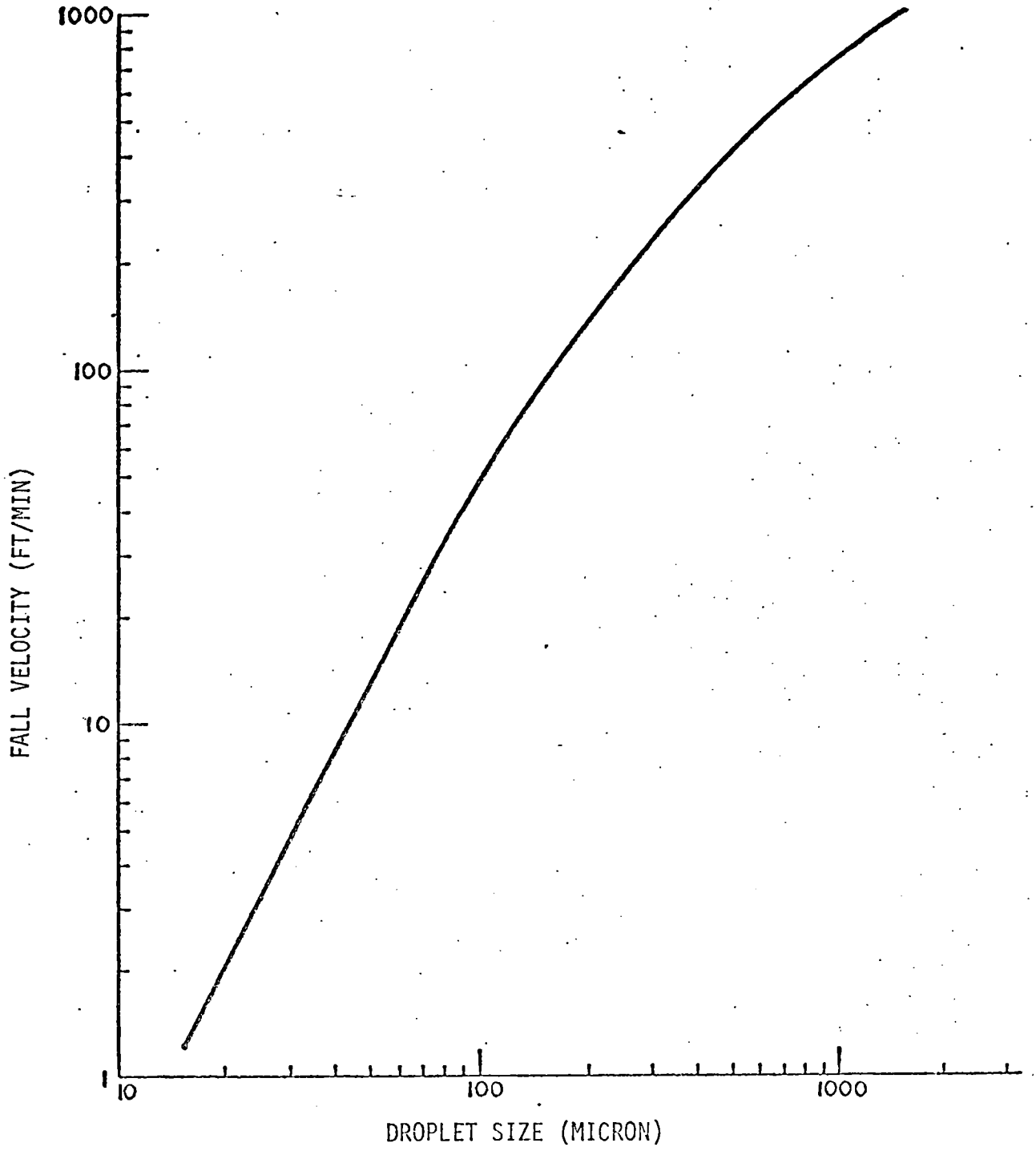


Figure 3-6. Fall velocity of water drops as function of size

It is seen in Fig. (3-6) that small droplets (say less than 100 microns) have extremely low velocity indicating a minor effect of droplet weight on their dynamic behavior. Their movement path is governed by aerodynamic forces such as wind and buoyancy of the exhaust plume. Plume buoyancy and atmospheric turbulence tend to keep these droplets in suspension for an extended period. These droplets will, in general, follow the plume path and their concentration at any point downwind will be ultimately calculated by atmospheric dispersion models (see the air-pollution study).

Fig. (3-6) indicates that large droplets (say around 2000 microns) have a high terminal fall velocity. Because of this reason, they fall very rapidly and most will hit the ground in the immediate vicinity of the tower. In this case their dynamic behavior is governed by gravity and wind induced drag forces. In other words, the movement of these droplets is essentially independent of the exhaust air plume path.

The majority of the droplets are in an environment experiencing partial or complete evaporation or condensation. These factors must be taken into account when the terminal fall velocities or droplet trajectories are to be calculated. The mathematical models prepared for these purposes are generally semi-empirical. The one developed by Ecodyne Corporation is a good example in this area.

As an illustration of what this computer program can do, an engineering case is included and presented below.

Inputs:

a eight-cell cross-flow tower

Circulating salt water flow	134,000 GPM
Drift rate	0.004%
Drift size distribution	See Table 6
Ambient dry-bulb temp.	63°F
Relative humidity	50%
Wind speed	20 MPH

The computer results are presented in Fig. (3-7). Results indicate that 68.7% of the drift mass hits the ground in the first 400 ft. About half of this total mass falls out in the first 150 feet. Drift originally smaller than 100 microns in diamter represents 12% of the total mass and for the atmospheric conditions selected, droplets originally smaller than 450 microns evaporate to a diameter of 100 microns or less before hitting the ground. Thus, 37.3% of the total drift mass remains in suspension and its subsequent behavior is evaluated using the atmospheric air-pollution model.

The dashed lines of Fig. (3-7) show the trajectory history when the atmospheric conditions are changed to 30°F dry-bulb temperature and 80% relative humidity. It is seen that the droplets fall slightly closer to the tower during the winter period.

In summary, it can be generally said that cooling tower drift effects on the environment are localized and that beyond some reasonable downwind distance that is usually within the plant site boundary, drift does not significantly affect the environment.

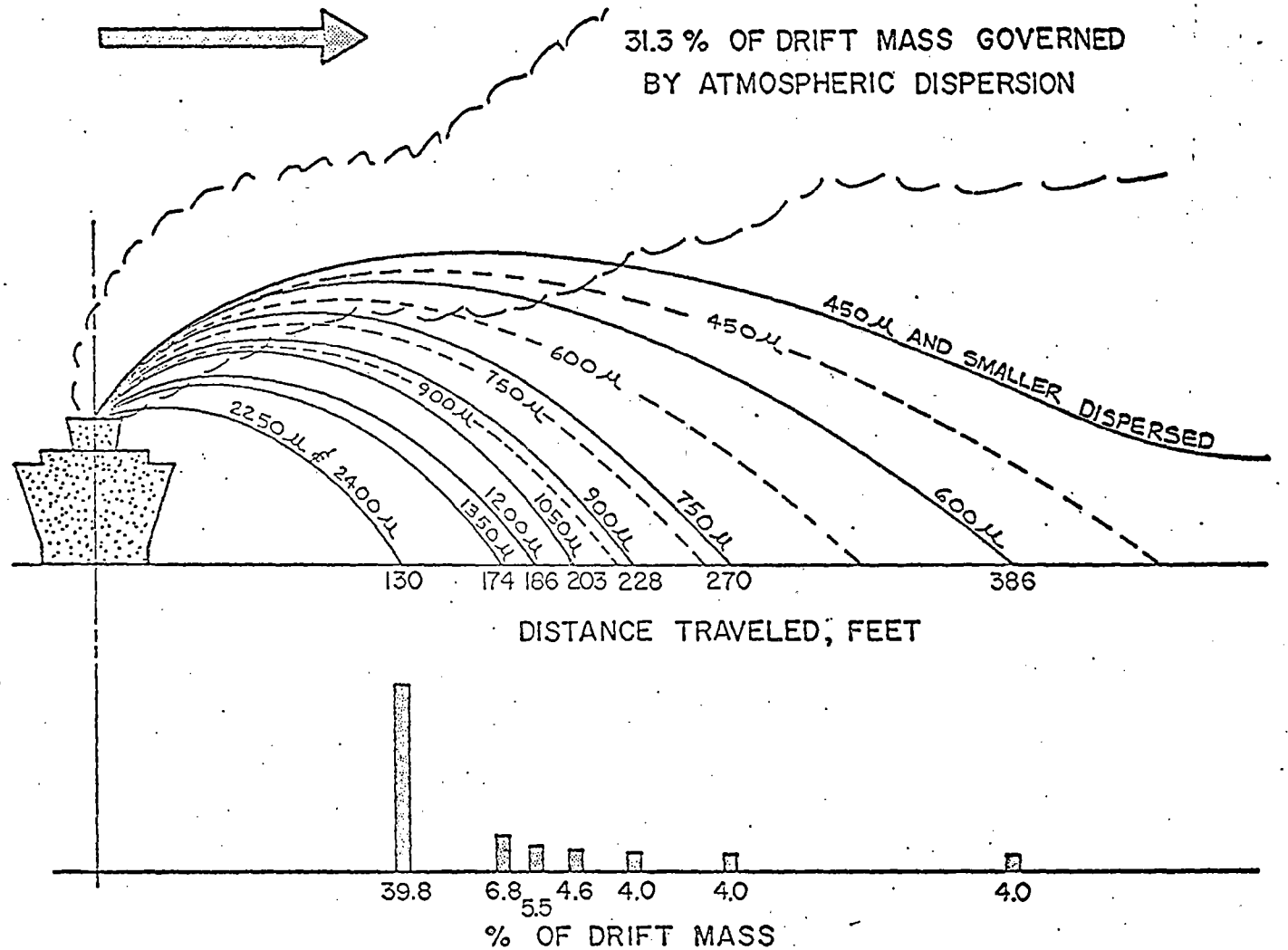


Figure 3-7. Drift dynamic behavior

3-4. Evaporation

In addition to drift, cooling towers emit to the atmosphere a large amount of water vapor which results from evaporation inside the tower. Evaporation is the main heat transfer mechanism in normal tower operation by which heat is transferred to the air. As water evaporates, heat is needed for evaporation (approximate 970 Btu is required for one pound of water evaporated). Because of this heat release, the water temperature is reduced.

It is of interest to know the amount of water vapor released during a tower operation. These data are also needed for cooling tower plume study. A quick estimate of water vapor release may be made by using the equation.

$$\dot{M}_e = C_1 \dot{Q}/TR \qquad (3\text{-}2)$$

where $\dot{M}_e$ = evaporation rate lb/hr

$\dot{Q}$ = heat load Btu/hr

TR = temperature rise °F

C_1 = empirical constant (0.02 - 0.04)

For example, a 680 MW fossil fuel power plant has a maximum heat load 3400×10^6 Btu/hr and temperature rise 30 F. Then, the evaporation rate will be in the range of 2.2×10^6 lb/hr to 4.4×10^6 lb/hr. It should be pointed out that the evaporation rate from cooling towers is not constant in operation. It is affected bythe weather conditions. In general, the evaporation would be reduced as the ambient temperature decreases, Considering the above example and using the evaporation rate

2.77×10^6 lb/hr (or 0.35×10^6 grams/sec), the cooling tower evaporation for various temperature ranges can be found and is indicated in Table 3-7.

It is seen that the evaporation rate decrease from 350,000 gram/sec to 200.000 grams/sec when the ambient air temperature drops from 23°C to -17°C. Associated with the decrease of evaporation, the heat transfer rate due to evaporation also drops from 210×10^6 cal/sec to 119×10^6 cal/sec. The balance is taken by convection heat transfer mechanism.

It is not difficult to understand the above physical phenomena. As the ambient air temperature decreases, the air has diminishing capacity to absorb the water vapor. In other words the saturated air always has less water content at low temperature than that at high temperature. As the evaporation process continuously takes place inside the tower, the concentration of dissolved solids in the remaining water would increase. Scale formation or corrosion in various components of the cooling system will eventually occur, depending upon the pH content in the water. Scale formation generally lowers the heat transfer ability of the system while corrosion leads to material failures. To maintain the proper pH and keep the concentration of dissolved solids below certain levels, a portion of the circulating water is bled off and discharged as waste. The discharged water is replaced with raw or treated water. This process is known as blowdown and may be performed continuously or intermittently.

It is of interest to know the amount of circulating water to be discharged. A quick estimate of blowdown may be made by the equation

Table 3-7

Effects of ambient air temperature

on the cooling tower evaporation rate

Ambient Air Temp. (°C)	Evaporation Rate (g/sec)	Latent heat (cal/sec)
$T > 23$	0.35×10^6	210×10^6
$5 < T \leq 23$	0.32×10^6	191×10^6
$-17 < T \leq 5$	0.25×10^6	153×10^6
$T \leq -17$	0.20×10^6	119×10^6

$$\text{Blowdown} = \frac{0.06 \times TR \times \dot{M}}{C-1} \qquad (3\text{-}3)$$

where Blowdown = blowdown water flow rate, gallons/hr

TR = temperature rise in condenser, °F

$\dot{M}$ = circulating water flow rate, gallons per minute

C = ratio of circulating water to raw water minerals concentration

Values of C run from 2 to 10. Take the previous example again where a 680 MW fossil-fuel power plant has a maximum heat load 3400 x 10^6 Btu/hr and temperature rise 30°F. Calculations will provide the circulating water flow rate 226,666 GPM. If C is selected to be 5, the blowdown is then

$$\text{Blowdown} = \frac{(0.06)(30)(266,666)}{5-1} \qquad (3\text{-}4)$$

$$= 102,000 \text{ gallons/hr}$$

This water requirement must, of course, be added to the amount of water losses due to evaporation, drift and other operations.

3-5. Formation of cooling tower fog.

One of environmental concerns due to a cooling tower operation is the modification to the local and downwind environment as the result of the ejection and diffusion of the cooling tower vapor effluent. Specifically, the concern is formation of cooling tower fog and frequency of its occurance. This section is intended to discuss the conditions for formation and characteristics of cooling tower fog.

Before physics of fog is discussed, it is important to understand how much water vapor content is needed to saturate the atmosphere. In the following is a numerical relationship between the water vapor required and ambient air temperature.

Temp. (°C)*	Vapor content required for saturation* g/m^3
-20	0.894
-10	2.158
0	4.847
+10	9.401
+20	17.300

*available from any standard text of thermodynamics (also see Fig. (3-8))

It is seen that as the ambient air temperature decreases, the air has a diminishing capacity to absorb the water vapor content. For a given temperature, the air would be in a supersaturated state if the water vapor content is larger than the saturation value. Since the air in the supersaturated state is not stable, part of the water vapor would be condensed into liquid droplets, and thus producing fog. In other words, formation of cooling tower fog will take place whenever the mixture of tower effluent and ambient air is a supersaturated state. This can be best described by reference to a diagram of temperature versus absolute humidity, as shown in Fig. (3-8). The heavy curbed line represents the required water vapor content for saturation of air. Point A in Fig. (3-8) is a saturated state which might represent the effluent from a cooling

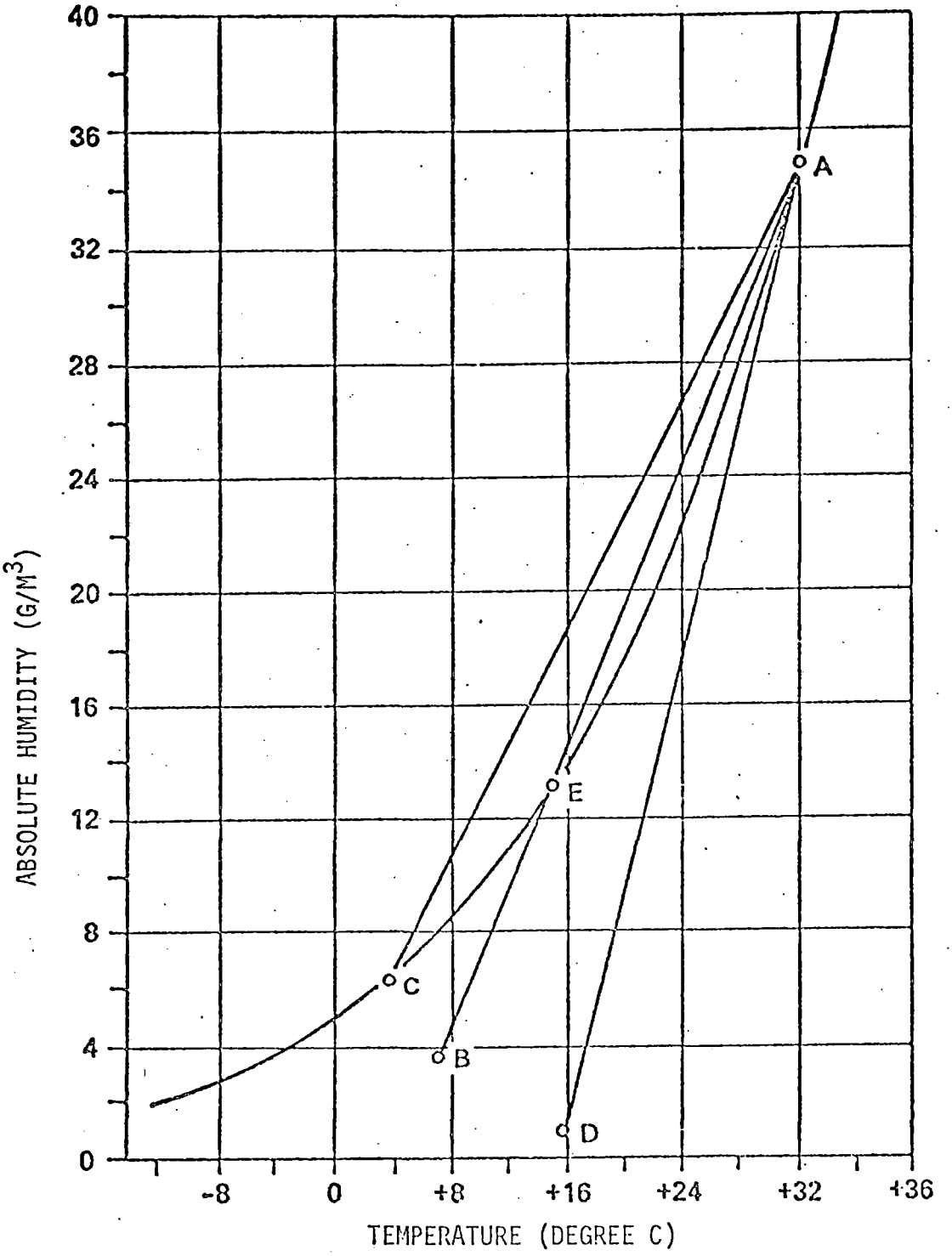


Figure 3-8. Saturation absolute humidity versus temperature

tower. Points B, C, and D are possible ambient atmospheric conditions. First, take the case that the atmosphere is at condition D, the mixture of tower effluent and ambient air is unsaturated in the entire process AD and thus produce no fog. On the other hand, if the ambient air is already saturated such as that indicated by point C, the mixture as seen in Fig. (3-8) is entirely in the supersaturated region and therefore produce the cooling tower Fog. Theoretically, the tower fog in this case would extend indefinitely. When the ambient condition is at B, the mixture is partly in the supersaturated region and partly in the unsaturated region. Those in the supersaturated region would produce cooling tower fog. The physical dimensions of the foggy plume depend upon the distance the plume travels from the cooling tower before it is diluted to the state E. At the state E the fog starts to evaporate.

To determine the frequency of cooling tower fog formation, we will define the saturation deficit q_d in the atmosphere. It is the difference between the saturation water vapor content q_s at a given temperature and the local ambient water vapor content q_e. Mathematically, that is

$$q_d = q_s - q_e \tag{3-5}$$

Evidently, the saturation deficit is closely related to climatical conditions at the plant site. Table 3-8 shows the typical distribution of saturation deficit in the northern part of the nation. It is

TABLE 3-8

Distribution of Saturation Deficit - Minneapolis
(based on 10 years of data)

Saturation Deficit	Hours/ Year	(%) Occurrence	(%) Cumulative
$< 0.10\ g/m^3$	55	0.6	0.6
0.11/0.20	438	5.1	5.7
0.21/0.30	403	4.6	10.3
0.31/0.40	606	6.9	17.2
0.41/0.50	378	4.3	21.5
0.51/0.75	967	11.0	32.5
0.76/1.00	791	9.0	41.5
1.01/1.50	624	7.2	48.7
1.51/2.00	559	6.4	55.1
2.01/3.00	923	10.5	65.6
3.01/4.00	536	6.1	71.7
4.01/5.00	455	5.2	76.9
> 5.00	2025	23.1	100.0
TOTAL	8760	100.0	

seen from the table that during more than 40% of the time of the year an addition of 1 gram of water per cubic meter of air would be sufficient to saturate the local air. Cooling tower effluant at release point contains from 20 to 40 gram/m^3 of water vapor above the ambient value. These concentrations are very quickly diluted, but nonetheless, a potential for fog formation clearly exists. In general, the fog formation would become especially serious in the winter time.

3.6 Cooling Tower Plume

The basic problem is to estimate the frequency and location of fog at the ground which would be produced by cooling towers. It is first to predict the dispersion of the cooling tower effluent, ie., the concentration in the atmosphere of heat and water vapor from the towers, as a function of position relative to the plant. Once the distribution of tower effluents is known for various weather conditions, the number of cases in which fog would be present is determined by reference to temperature and humidity statistic.

The computerized version of the model used for this purpose is similar to that for the air pollution study. That is the Pasquill-Gifford diffusion model. For reference, the basic equation of the method is presented as below

$$X\,(x,o,o,H) = \frac{Q}{\pi\,\sigma_y\,\sigma_z\,u}\quad \exp\left[-\frac{1}{2}\left(\frac{H}{\sigma_z}\right)^2\right] \qquad (3\text{-}6)$$

where

$X(x,o,o,H)$ = The concentration of released substance (either water vapor or heat in this study) in the atmosphere at distance x downwind from the source at the centerline of the plume (y=o) and at the ground level (z=o). The release is assumed to take place at an effective height H. above the ground

$\dot{Q}$ = the rate of substance release

σ_z, σ_y = the standard deviations of the distribution of effluent in the cross-wind (y) and vertical (z) direction. σ_z and σ_y are function of x and meteorological conditions

u = wind speed

The coordinate system showing this distributions is indicated in Fig. (3-9). In the following is a summary of calculational procedures used in this kind of analysis.

1. use the plant site information and weather data to prepare a combined probability table for wind speed and stability.

2. use the weather and engineering data to determine the effective height. The effective source height is defined as a sum of stack height and rise of vapor plume.

3. use the engineering data to determine the source strength and configuration. These include the rate of substance release (heat and water vapor) which is a function of ambient air temperature and power plant loadings.

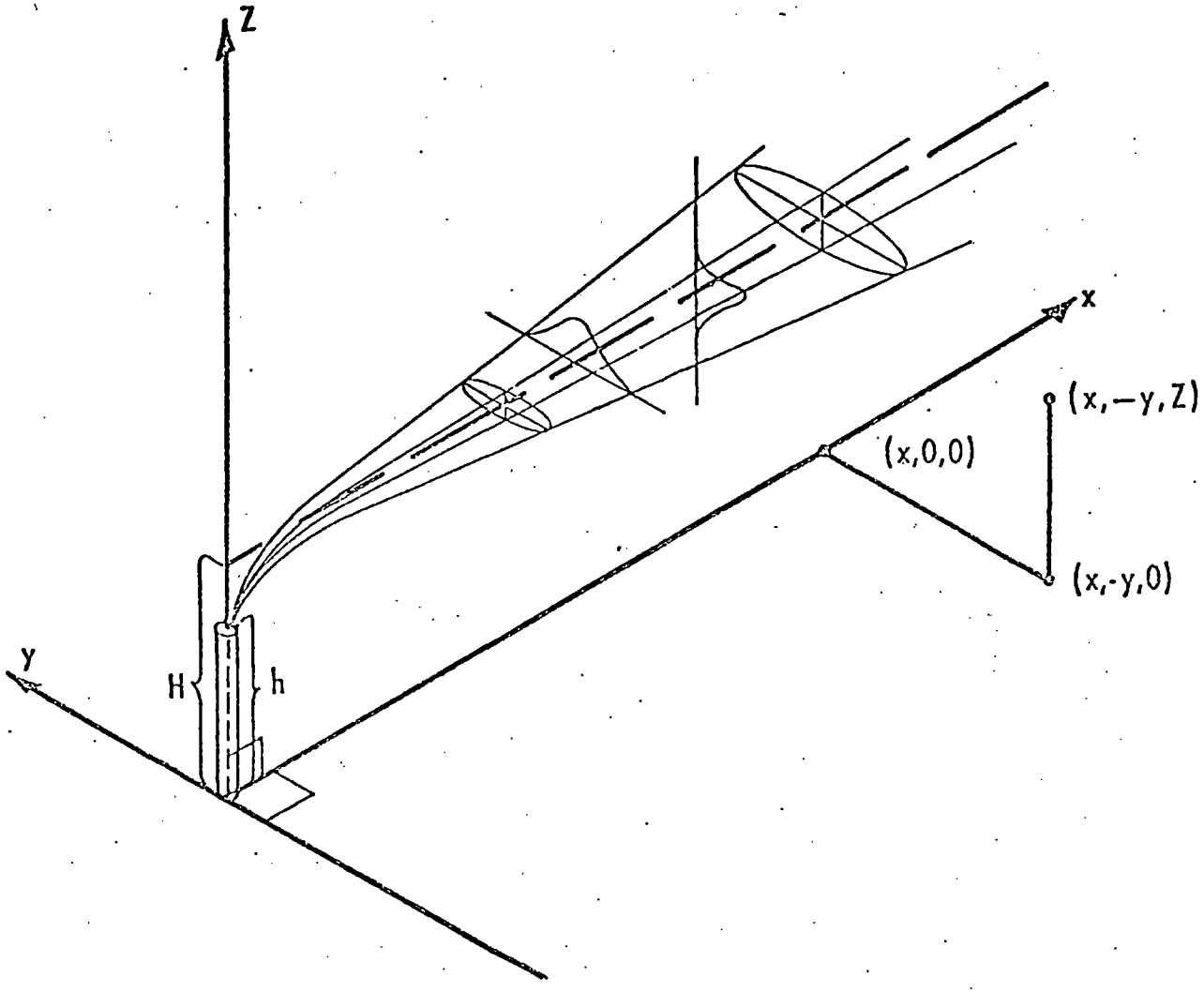


Figure 3-9. Coordinate system showing Gaussian distributions in the horizontal and vertical

4. use the mathematical model to calculate water vapor concentration and temperature rise at the downwind distance under various conditions.

5. add the calculated values to the ambient values (temperature rise and water vapor concentration) to determine whether or not the moisture exceeds the saturation value at a given temperature. If the state is saturated or supersaturated, fog is assumed to occur. If the state of the mixture (tower effluent and ambient air) is unsaturated, no fog is considered to exist.

As presented above, the calculational procedures are quite straight forward. To illustrate them, we present an engineering case study as follows:

A power Company planned to construct and operate a 1,360,000 kilowatt, coal-fired electric generating plant in the state of Minnesota. The plant would include two 680,000 kilowatts generating units. It was decided that mechanical-draft cooling towers were used to discharge thermal effluents. [64]

As step No. 1, the combined probability table for wind speed and stability was prepared using the local weather data information. It is shown in Table 3-9 and Table 3-10 . These tables would give the percentage occurrence of various wind speeds and directions for each of four stability categories.

In Step No. 2, the effective source heights were calculated. Results were shown in Table 3-11 . As it was indicated in the air pollution study, the effective height is a function of prevailing

TABLE 3-9

Joint Probability - Stability and Wind Speed (%)

Wind Speed (mph)	STABILITY			
	Unstable	Neutral	Slightly Stable	Stable
1-4	0.8	0.3	0.3	0.9
5-14	13.7	7.4	7.0	17.9
15-24	16.8	11.1	11.5	8.3
> 25	1.9	0.8	0.4	0.1

TABLE 3-10

Conditional Wind Direction Probabilities (%)

Wind 1-4 (mph)	NNE	NE	ENE	E	ESE	SE	SSE	S	SSW	SW	WSW	W	WNW	NW	NNW	N
Unstable	9	0	4	4	2	4	9	5	4	13	4	13	4	7	9	7
Neutral	0	16	0	11	15	6	15	11	5	6	0	6	0	1	5	1
Sl. Stable	9	5	5	14	9	14	0	9	9	1	0	1	5	9	0	14
Stable	7	3	5	3	8	5	7	15	7	5	11	5	7	1	8	5
Wind 5-14																
Unstable	5	5	8	3	4	4	5	5	11	7	6	5	8	8	12	6
Neutral	5	5	11	6	13	8	11	4	4	3	3	2	4	6	13	6
Sl. Stable	4	5	5	4	5	7	8	7	7	3	4	5	7	7	15	6
Stable	2	2	3	3	5	5	9	11	10	8	8	5	10	9	8	3
Wind 15-24																
Unstable	3	4	4	2	7	5	6	8	9	4	4	5	12	8	13	7
Neutral	5	3	8	3	7	5	11	6	5	4	2	2	7	13	15	4
Sl. Stable	3	2	5	3	5	6	9	5	10	5	4	3	9	16	13	3
Stable	1	1	1	1	3	5	8	11	13	8	6	9	8	10	10	4
Wind >25																
Unstable	1	0	1	1	2	1	2	8	11	0	1	3	13	30	22	1
Neutral	0	2	4	2	4	0	2	6	4	11	0	0	20	33	14	0
Sl. Stable	0	0	0	0	4	0	0	38	15	7	0	0	4	31	0	0
Stable	0	0	0	0	0	0	0	50	25	25	0	0	0	0	0	0

TABLE 3-11

Effective Source Heights (Meters) for Use in Dispersion Calculations

STABILITY	WIND SPEED 1 m/sec	4.3 m/sec	8.6 m/sec	15 m/sec
Unstable	250	200	150	100
Neutral	250	200	150	100
Slightly Stable	150	100	50	40
Stable	100	70	40	30

meteorological conditions as well as the discharge velocity and temperature. The choice of plume rise equations is also important for it significantly affects the calculated ground-level concentrations. The values shown in Table 3-11 were considered fairly conservative. In other word, they were generally smaller than the observed value and therefore, would lead to higher calculated ground-level concentrations.

As step No. 3,the source strength was prepared using the engineering data provided by cooling tower manufactures. These were presented in Table 3-12 and Table 3-13. It is seen that the source strength is also related to the ambient air temperature. When the ambient air temperature decreases, the emission rate of water vapor would decrease and at the same time, the emission rate of sensible heat would increase. In this engineering case study, the plant was assumed to operate at a full load. That is, the heat load of cooling towers (the sum of latent heat and sensible heat releases) would remain unchanged.

To use the mathematical model, two things were needed. They were the discription of virtual point source and the standard deviations of the distribution of effluent σ_y and σ_z. The virtual point source was used to approximate the emitting area of cooling towers which in this case study was 400 x 1000 square feet (there were two virtual point sources, each for one generating unit). The virtual point source would provide a hypothetical cloud of moist air at the actual location of the cooling towers. Then, this

TABLE 3-12

Cooling Tower Parameters

Air Flow Per Cell	1.5×10^6 cfm
Outlet Velocity	1,800 fpm
Outlet Diameter	32 ft
Height of Outlet	60 ft
Heat Load	3.4×10^9 BTU/hr per unit

TABLE 3-13

Cooling Tower Output Conditions for Various
Ambient Temperature Ranges
(Source Strengths for Diffusion Model)

Range of Ambient Temperature (deg C)	Evaporation (g/sec)	Latent Heat (cal/sec)	Sensible Heat Released (cal/sec)
	(a) 28 cells		
$T > 23$	7.0×10^5	42.1×10^7	5.4×10^7
$5 < T \leq 23$	6.4×10^5	38.2×10^7	9.3×10^7
$-17 < T \leq +5$	5.1×10^5	30.6×10^7	16.9×10^7
$T \leq -17$	4.0×10^5	23.8×10^7	23.7×10^7
	(b) 40 cells		
$T > 23$	6.9×10^5	41.4×10^7	6.1×10^7
$5 < T \leq +23$	6.2×10^5	37.2×10^7	10.3×10^7
$-17 < T \leq +5$	4.4×10^5	26.7×10^7	20.8×10^7
$T \leq -17$	2.9×10^5	17.4×10^7	30.1×10^7

All numbers are totals for two generating units at full load.

cloud of moist air experienced a Gaussian distribution in the downwind distance. The distances between the virtual point source and real source were calculated and shown in Table 3-14. These distance were required in the model applications.

The standard deviations σ_y, σ_z were function of downwind distance and meteorological conditions. Fig. (3-10) and Fig. (3-11) showed the dispersion coefficients at a downwind distance from the virtual point source. At the location of the cooling towers the dispersion coefficients σ_y, σ_z were estimated and presented in Table 3-14.

In this engineering case study the computer model was used to determine the concentration of water vapor and the temperature rise. These values were added to the ambient values and then, decision was made whether or not the cooling tower fog would exist. Table 3-15 presented the results of computer calculations. The numbers in Table 3-15 represent average hours per year of cooling tower fog without regard to direction. The directional distribution of cooling tower fog is shown in Fig. (3-12) and Fig. (3-13). Fig. (3-12) gives a detailed analysis of the directional distribution within 4 km of the proposed plant site. Fig. (3-13) shows smooth contours out to 20 km distance.

The results indicate that fog should be expected in the town of Becker on a maximum of 15 hours per year, on highway 10 up to 24 hours per year, on Highway 152 and proposed Highway I 94 up to 20 hours per year.

TABLE 3-14

Parameters of Initial Source Description for Cooling Towers

Stability (Turner[7])	x_v (meters)	σ_y (meters)	σ_z (meters)
A	430	100	80
B	610	100	63
C	950	100	58
D	1500	100	43
E	2100	100	35
F	3300	100	28

x_v = distance upwind to imaginary point source

σ_y, σ_z = standard deviation of Gaussian distribution of effluent in y (crosswind) and z (vertical) directions at cooling tower complex. Size of cloud is approximately 4.3 σ in each direction.

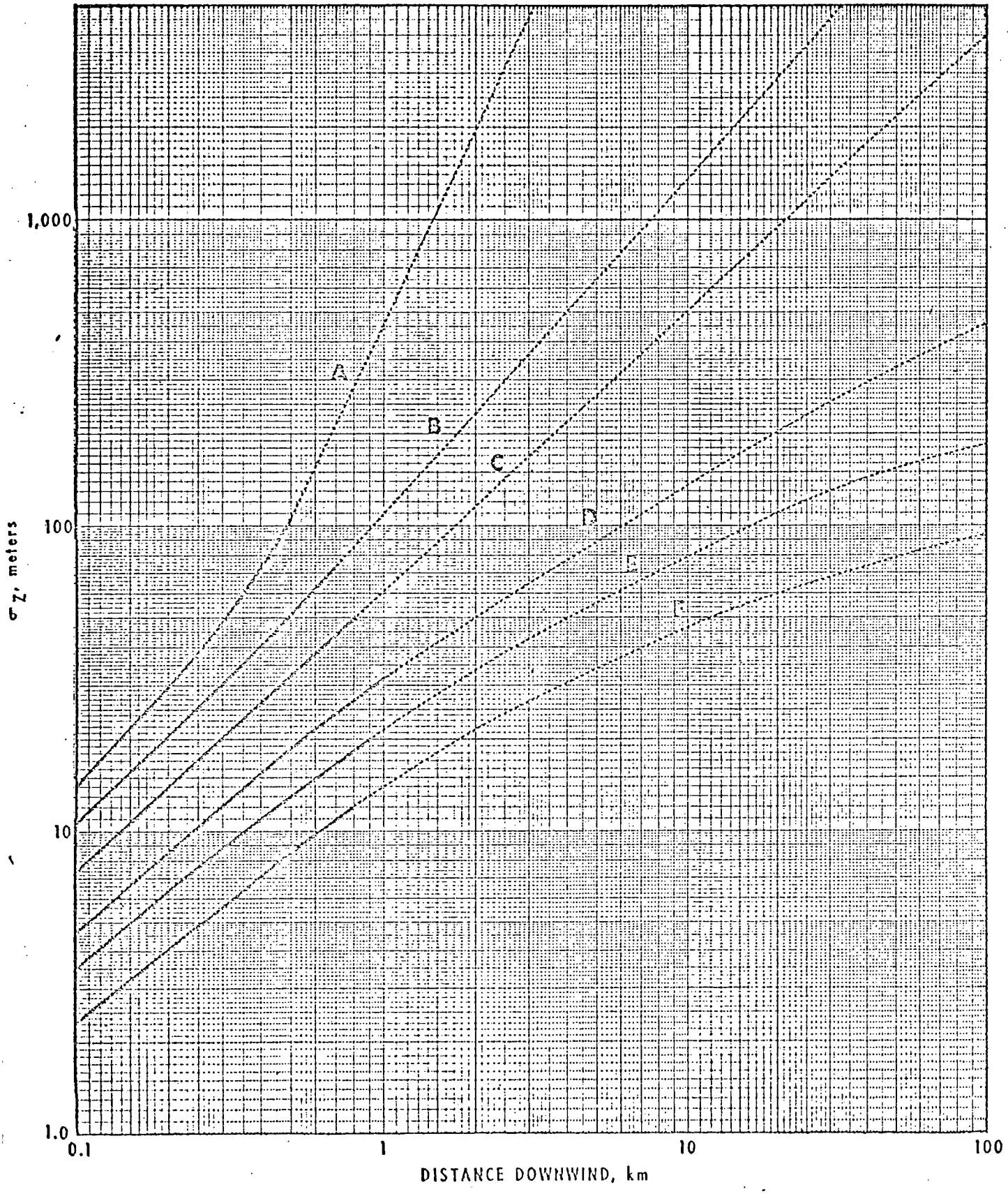


Figure 3-10. Vertical dispersion coefficient as a function of downwind distance from the source

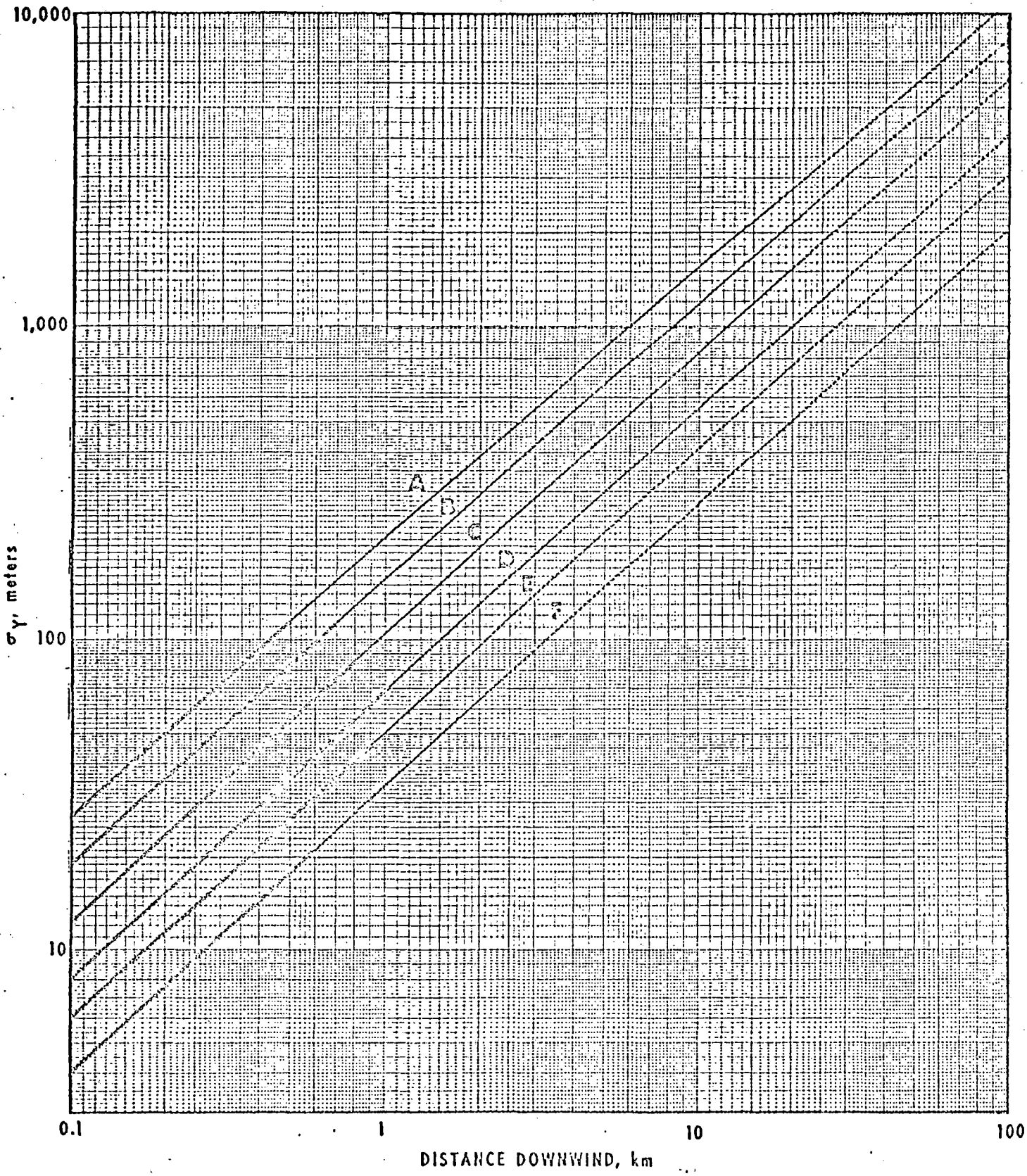


Figure 3-11. Horizontal dispersion coefficent as a function of downwind distance from the source

TABLE 3-15

Estimated Fog Frequency
(Hours per Year)

	0.4 km (1270 ft)	2 km (1.24 mi)	4 km (2.5 mi)	20 km (12.4 mi)
(1) Original tower size (14 cells) and spacing	449	607	515	348
(2) Towers with 20 cells per unit. Same spacing as original design	381	516	438	286
(3) Original towers in two groups with 3000' baseline	245	250	229	348
(4) Low-efficiency towers (as in 2) separated (as in 3)	214	219	209	286

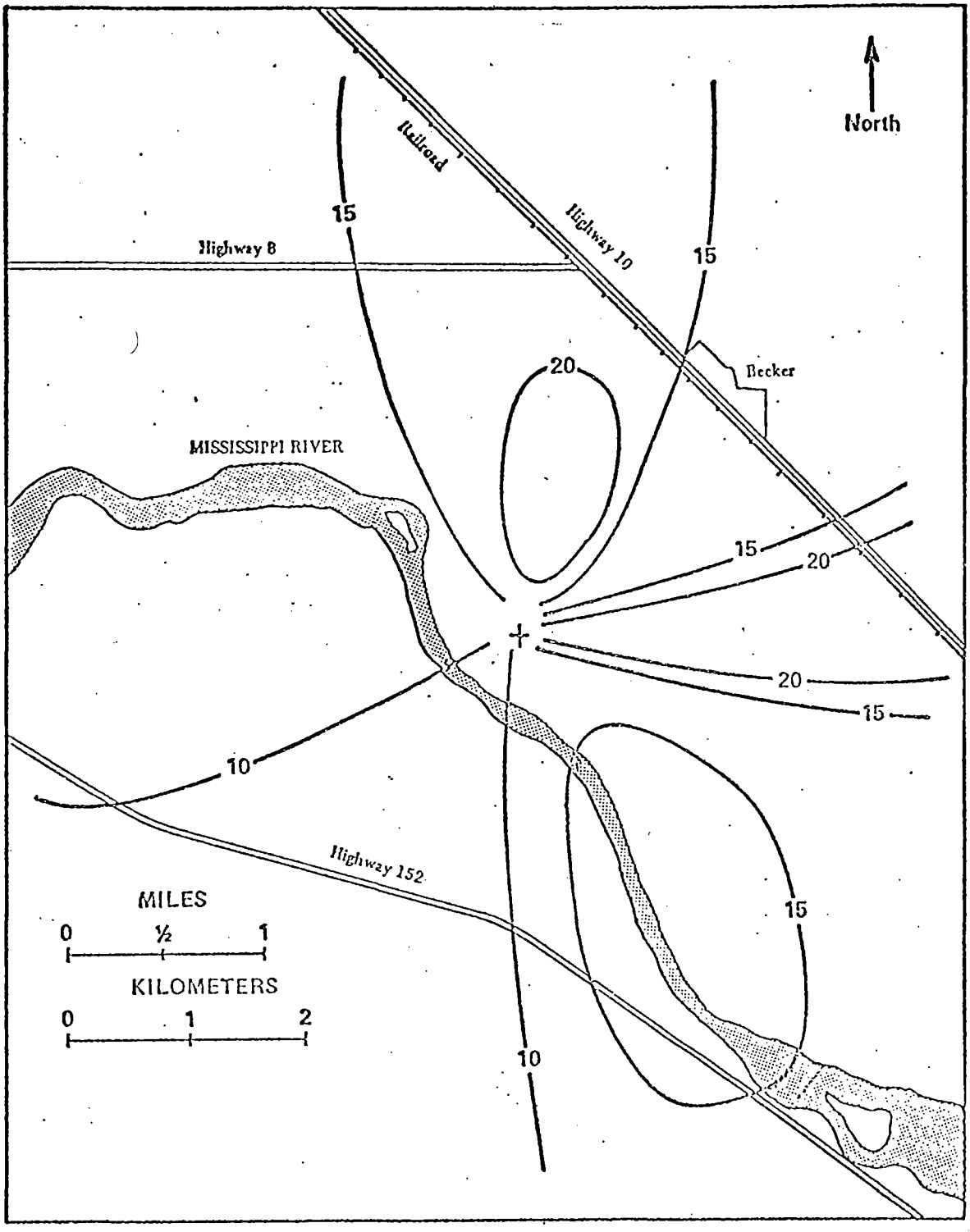


Figure 3-12. Contours of predicted fog frequency within 2 miles of cooling towers (20-cell towers in two seperated groups)

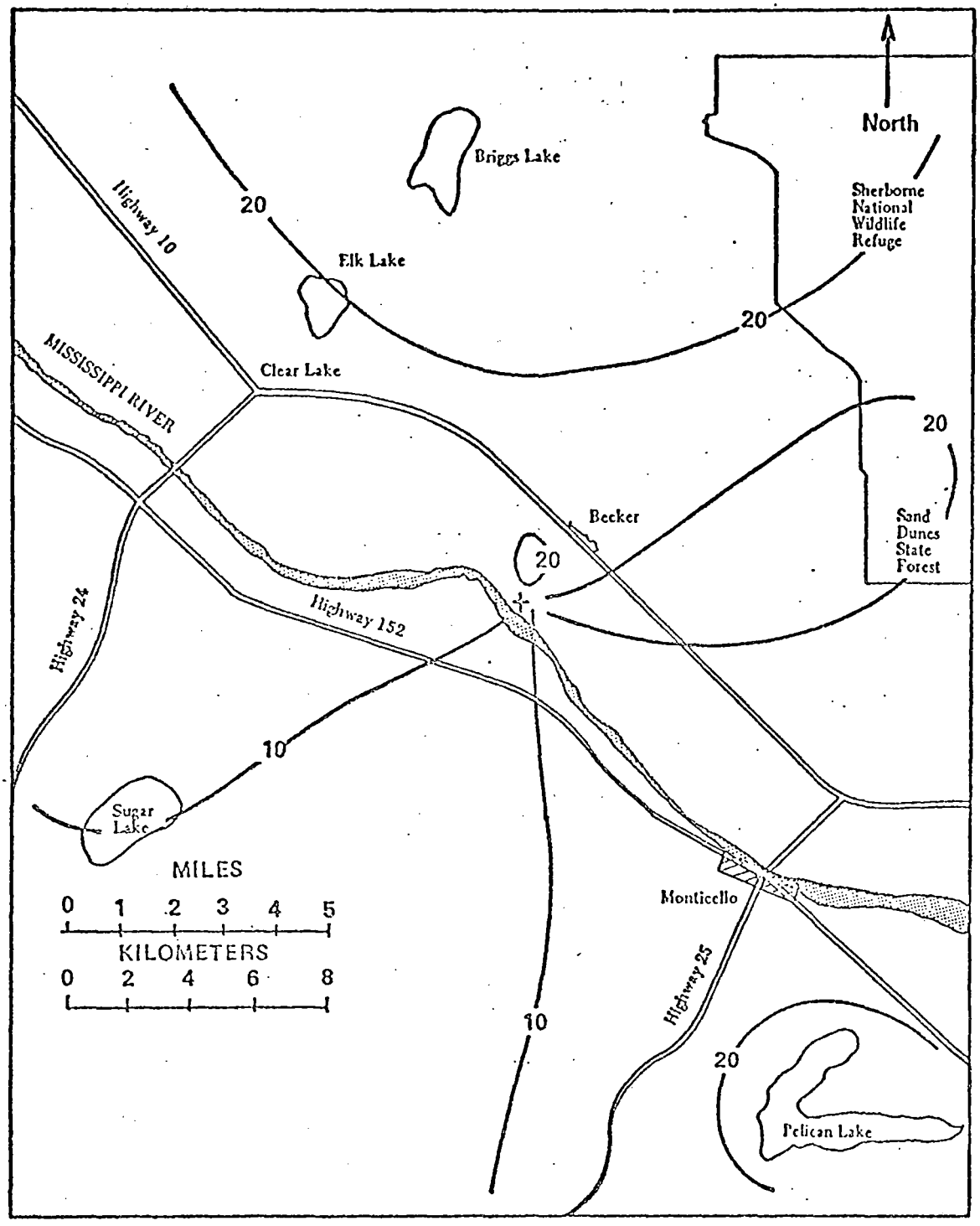


Figure 3-13. Smoothed pattern of predicted fog frequency around cooling towers (20-cell towers in two seperated groups)

It is important to know the extent to which the calculated cooling tower fog hours would coincide with hours during which natural fog would already be present. In this case study it was found that 35% of the calculated fog hours are times when the ambient humidity is 90% or greater. Since natural fog is usually found in the atmosphere when the relative humidity exceeds 90%, it is estimated that about 35% of the cooling tower fog cases would coincide with natural visibility reduction.

Before this section in concluded, it is important to point out that the above-mentioned procedures are only valid for the vapor plume emitted from mechanical-draft cooling tower. The plumes from natural-draft towers should be treated differently. Because of climatic conditions, it is not expected to have natural-draft tower operation in the State of North Dakota and therefore it is omitted from this study.

4. COOLING POND

Cooling ponds are increasingly used as a source of circulating water for power plant operation. This is especially true when land is available at a reasonable price and where geograpic conditions are suitable for pond construction. In general, cooling ponds are the simplest, cheapest and least water intensive method of water cooling. Also, this cooling method makes it possible to develop the lake and the power plant near the fuel source or near the system load center. However, cooling ponds have low heat transfer rates, thereby requiring large pond surface areas. They also concentrate dissolved solids, and collect impurities form the atmosphere, thereby requiring frequent blowdowns.

This section is intended to cover the fundamentals of cooling pond for power plant operations. The possible environmental impacts due to a cooling pond operation are presented. This section also includes a case study showing the relationship between the design parameters and impacts of cooling pond on its environment.

4-1. Fundamentals of Cooling Pond.

Cooling pond is a man-made water body in which the circulating water will disscipate its heat into atmosphere and reduce its temperature. The heat transfer mechanisms between the water surface and atmosphere are complicated. They involve radiation, convection, conduction and evaporation. The components of heat transfer rate on the water surface may be written as follows:

$$\phi_n = \phi_s - \phi_{sr} + \phi_a - \phi_{ar} - \phi_{br} - \phi_e - \phi_c \qquad (4\text{-}1)$$

where ϕ_n = net heat input to a water body.

ϕ_s = short-wave radiation incident to the water surface.

ϕ_{sr} = reflected short-wave radiation.

ϕ_a = incoming long-wave radiation for the atmosphere.

ϕ_{ar} = reflected long-wave radiations.

ϕ_{br} = long-wave back radiation emitted by the body of water.

ϕ_e = energy utilized by evaporation.

ϕ_c = energy conducted from the body of water as sensible heat.

The above equation is valid under steady-state conditions. If all the incident and reflected radition terms are grouped in a net radiation term ϕ_r, we have then

$$\phi_n = \phi_r - (\phi_{br} + \phi_e + \phi_c) \qquad (4\text{-}2)$$

Evidently, ϕ_r is a function of meterological variables and the quantity $\phi_{br} + \phi_e + \phi_c$ depends in part on the water surface temperature. There have been many works done for predication of these terms. The following equations are frequently used:

$$\phi_{br} = 4 \times 10^{-8} (T_s + 460)^4 \qquad (4\text{-}3)$$

$$\phi_e = (68{,}000 - 34\ T_s)\ f(w)\ (e_s - e_a) \qquad (4\text{-}4)$$

and $$\phi_c = (68{,}000 - 34\ T_s)\ f(w)\ (0.0102(T_s - T_a)) \qquad (4\text{-}5)$$

where T_s = water surface temperature.

T_a = ambient air temperature.

e_a = vapor pressure of water at the ambient air

e_s = saturation vapor pressure of water at temperature. T_s.

$f(w)$ = wind empirical function

The units used in these equations are Btu/ft^2/day. The radiation term ϕ_r is commonly obtained by field measurements. It should be noted the evaporation terms are only valid for $e_s > e_a$. That is when the evaporative heat transfer is out of water. During the time period when $e_s < e_a$, the evaporative term should be set equal to zero.

In cooling pond desgin two important parameters are frequently used. They are the equilibrium temperature and surface heat exchange coefficient. The equilibrium temperature is the water surface temperature which, for a given set of meteorological conditions, makes the back radiation, evaporation and conduction losses exactly equal to the radiation inputs, that is, the net heat exchange ϕ_n is zero. The surface heat exchange coefficient is defined to give the incremental change of net heat exchange induced by an incremental change of water surface temperature. With these two concepts, the basic equation for cooling pond design is simply written as follows:

$$Q = KA\ (T_s - E) \qquad (4\text{-}6)$$

where: Q = heat rejection from cooling pond. Btu/day.

A = water surface area ft^2.

K = surface heat exchange coefficient Btu/ft^2/day/°F.

T_s = water surface temperature °F.

E = equilibrium temperature °F.

The derivations of expressions for calculation of E and K are lengthy, but not difficult to understand. For the purpose of this report they are simply summarized as follows:

$$T = (T_s + T_d)/2 \tag{4-7}$$

$$\beta = 0.255 - 0.0085T + 0.000204T^2 \tag{4-8}$$

$$f(u) = 70 + 0.7u^2 \tag{4-9}$$

$$K = 15.7 + (\beta + 0.26)\ f(u) \tag{4-10}$$

$$E = T_d - H_s/K \tag{4-11}$$

where: T_d = dew point temperature.

$f(u)$ = wind velocity function.

H_s = gross solar radiation. Btu/ft^2/day.

It should be pointed out that other expressions are available for the terms E and K. The discrepancies are due to different mathematical modelings from which the equations are obtained. It is these facts that make a cooling pond design much more than a routine engineering job.

Equations (4-7) through (4-11) are useful in predicting the average surface temperature of a cooling pond. The step by step procedures are iterative in nature and summarized as follows.

1. Assume the average water surface temperature. T_s.
2. Obtain the dew point temperature T_d from the site weather date.
3. Calculate the average temperature $(\frac{Ts + Td}{2})$.
4. Calculate the parameter β.
5. Calculate the wind velocity function.
6. Determine the surface heat exchange coefficient.
7. Use the site information to prepare the gross solar radiation H_s.
8. Determine the equilibrium temperature E.
9. Use the basic equation and the plant heat load per unit area to calculate the average water surface temperature.
10. Check the average water surface temperature with that assumed in step 1.

The calculational process is completed only when the assumed and calculated values lie within a certain range. In theory there is always one average water surface temperature for a given set of site conditions and plant heat load per unit pond area. In practice the cooling pond must be so designed that it would dissipate the heat rejected by the power plant and at the same time yield the lowest possible intake temperature of the cooling water. These dual design objectives are not always met. The intake water temperature is always of interest to engineers as well as environmentalists. Generally,it depends upon the pond hydrodynamic structure and its circulation pattern. For simplicity,

this report presents two kinds of cooling ponds: (a) completely mixed ponds and (b) flow-through ponds.

In a completely mixed pond the flow between the intake and discharge combined with wind moving tends to keep the pond at a nearly uniform temperature. The drop in temperature from the plant discharge to the pond temperature takes place in a small portion of the pond. Because of these facts, the intake water temperature is generally taken as the average pond temperature which can be calculated by:

$$\frac{T - E}{T_o - E} = \frac{1}{1 + \gamma_1} \tag{4-12}$$

where

$$\gamma_1 = \frac{KA}{\rho C_p Q_p} \tag{4-13}$$

In a flow-through pond the temperature decreases continually along the pond. The relationship between the pond temperature and surface area is

$$\frac{T - E}{T_o - E} = e^{-\gamma_2} \tag{4-14}$$

again

$$\gamma_2 = \frac{KA}{\rho C_p Q_p} \tag{4-15}$$

It is seen in Eq (4-14) that the temperature at a pond is a function of the pond area measured from the discharge point. The equation is strictly valid for a long narrow pond with no natural or vertical temperature gradient. The nomenclature of Eq (4-12) and Eq (4-14) are

T = temperature at a pond

T_o = condenser discharge temperature

E = equilibrium temperature

K = surface heat exchange coefficient

A = pond surface area

ρ = water density

C_p = specific heat of water

Q_p = condenser discharge volume flow rate

4-2. Parameters affecting the pond size.

The cooling pond must have sufficient surface area to cool the water to a temperature that will insure a satisfactory operation of the power plant. The fundamental equations presented in last section are frequently used to size the cooling pond for a given set of operation conditions or to predict the pond intake temperature for a given pond size.

A quick estimate of the size of a cooling pond may be made by using the equation

$$Q = hA\Delta T \qquad (4\text{-}16)$$

where Q = heat rejection to the pond.

ΔT = temperature difference between the air and average water temperature.

h = heat transfer coefficient with a typical value 3.5 Btu/hr/ft^2/F.

Another "rule of thumb" sizing is 1 acre/MW + 20% for surrounding land.

It whould be pointed out that the cooling pond design is not just to calculate the required pond size, but to select the optimal size so that the generation cost would become minimum. There are many parameters which would affect this optimal selection. In the following these parameters will be identified and investigated qualitatively through an illustrative example.

For purposes of this study, a 2400 psig, 1000°F/1000°F fossil fuel-fired plant of 2200 MW was assumed as a reference case. This plant was assumed to be located either in the northern half or in the southern half of the United State with the turbine being back end loaded at maximum capability conditions. The fixed charge rate for this plant was to be 18%, the fuel cost 0.30 dollar/MBtu, the load factor 75% and the average lake cost a 2,500 dollar/acre. It is noted that the lake cost not only must consider the cost of the land but must consider the costs to develop the site. Under these conditions it was found that the optimal lake size as shown in Fig. (4-1) is 1800 acres for Northern plant or 1900 acres for southern plant.

Using the reference plant and associated economic parameters, a sensitivity study was conducted. The results were presented in Fig. (4-2) to Fig. (4-6). These curves reflect the effects on the optimum lake size due to changes in the capability charge, lake cost, fuel cost, fixed charge rate and load factor. It can be seen from Fig. (4-2) to Fig. (4-6) that changes in the capability charge and lake cost have the greatest effect on the optimum lake size while changes

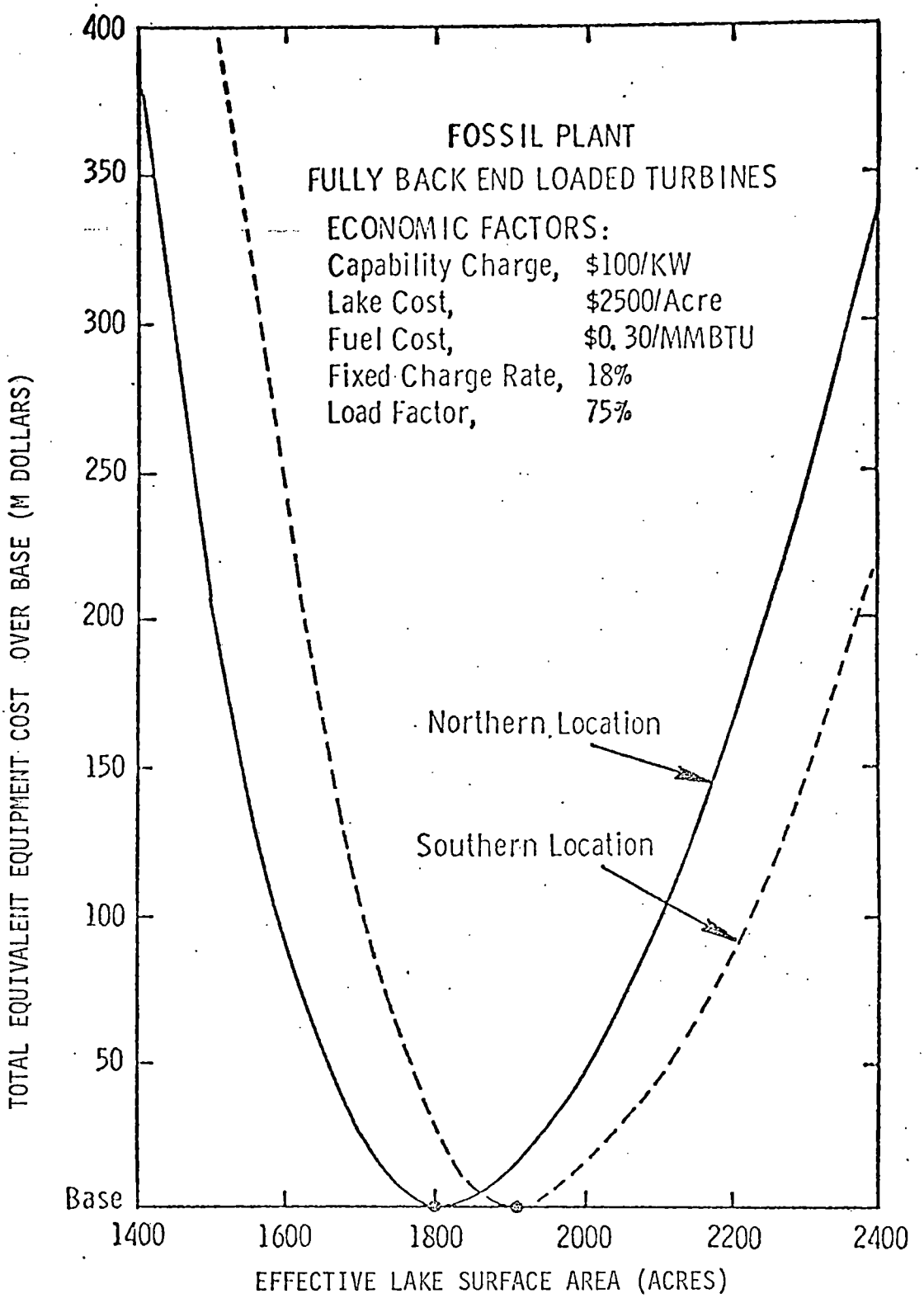


Figure 4-1. Equivalent equipment cost versus lake surface area

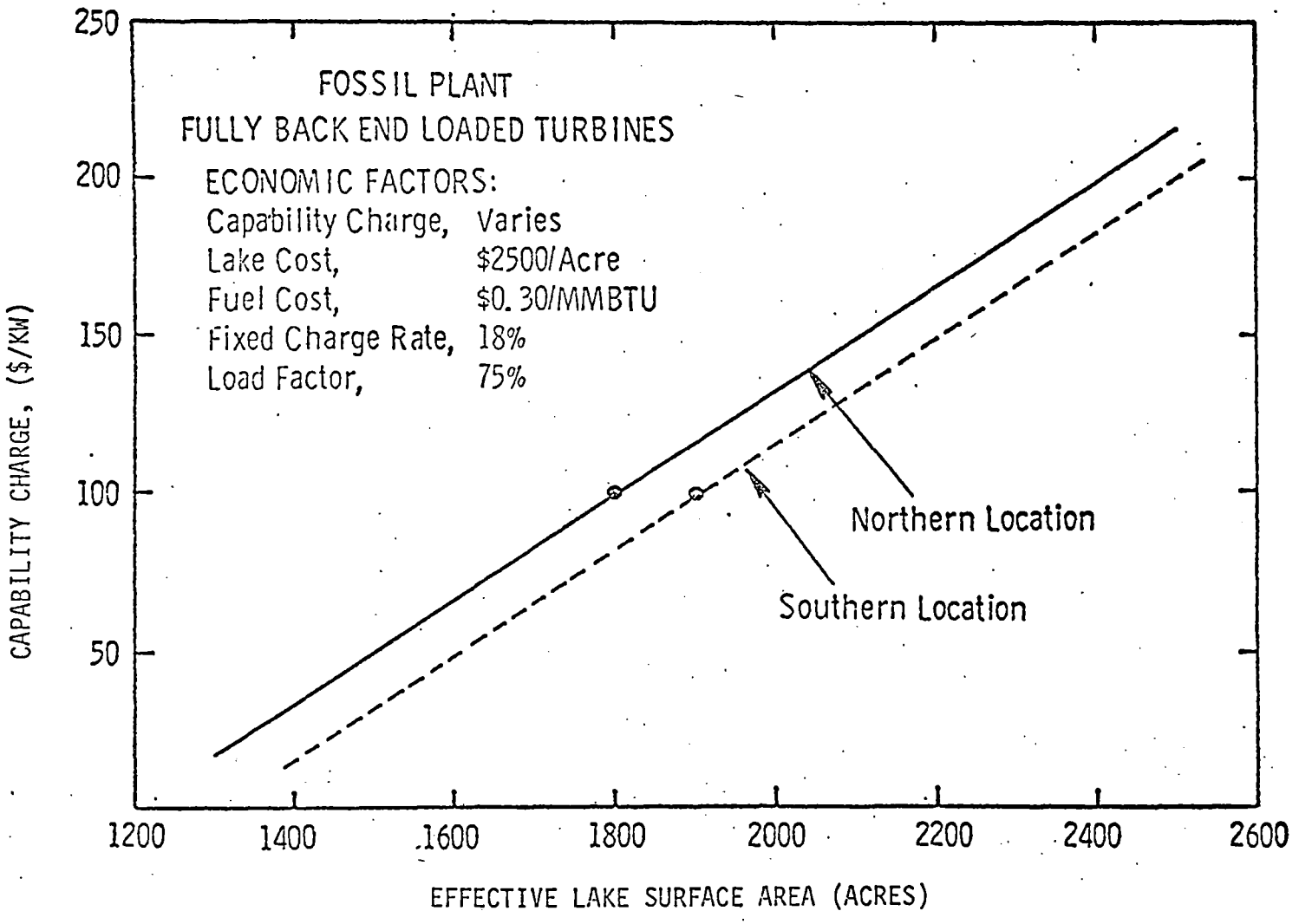


Figure 4-2. Capability charge versus lake surface area

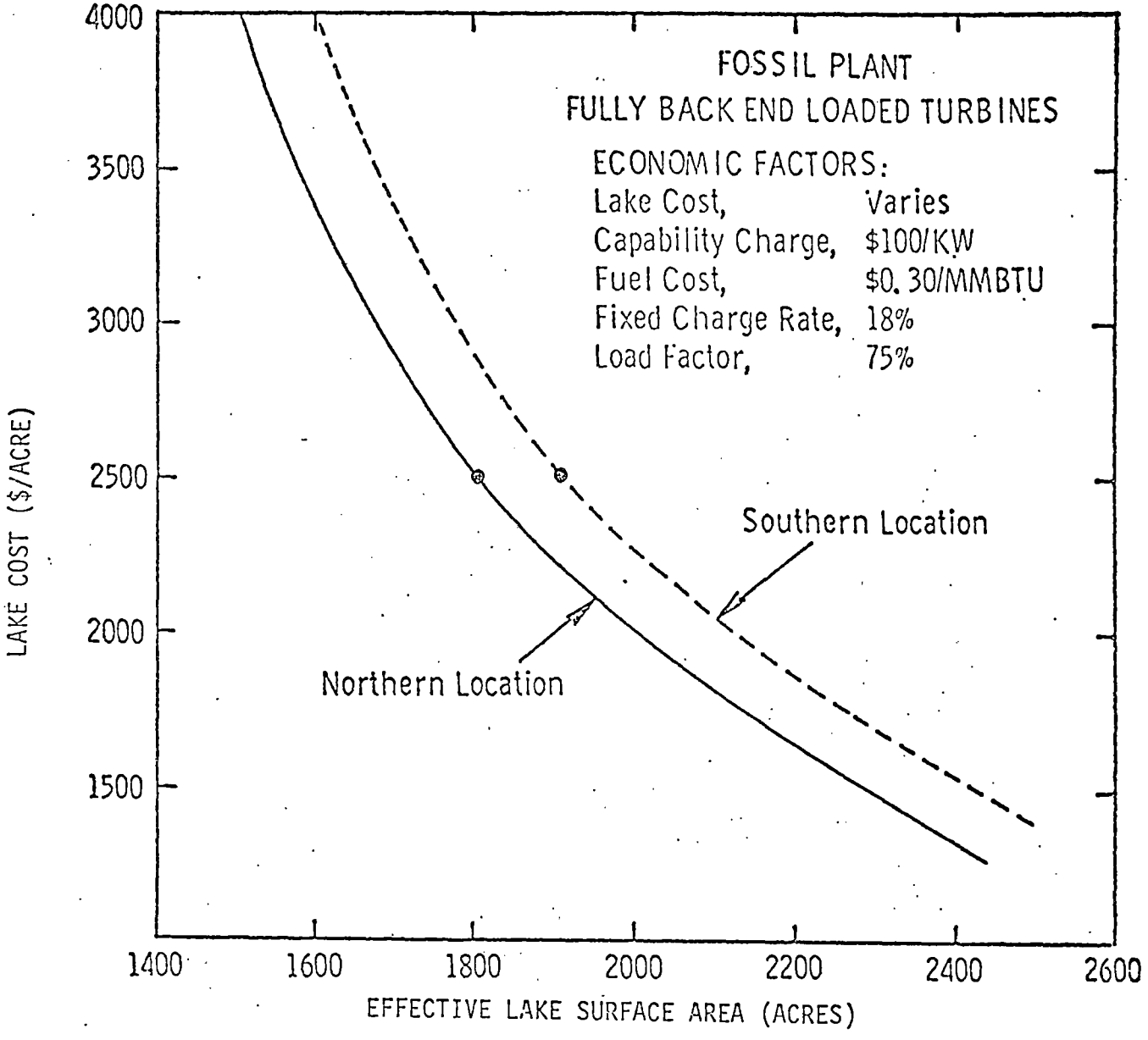


Figure 4-3. Lake cost versus lake surface area

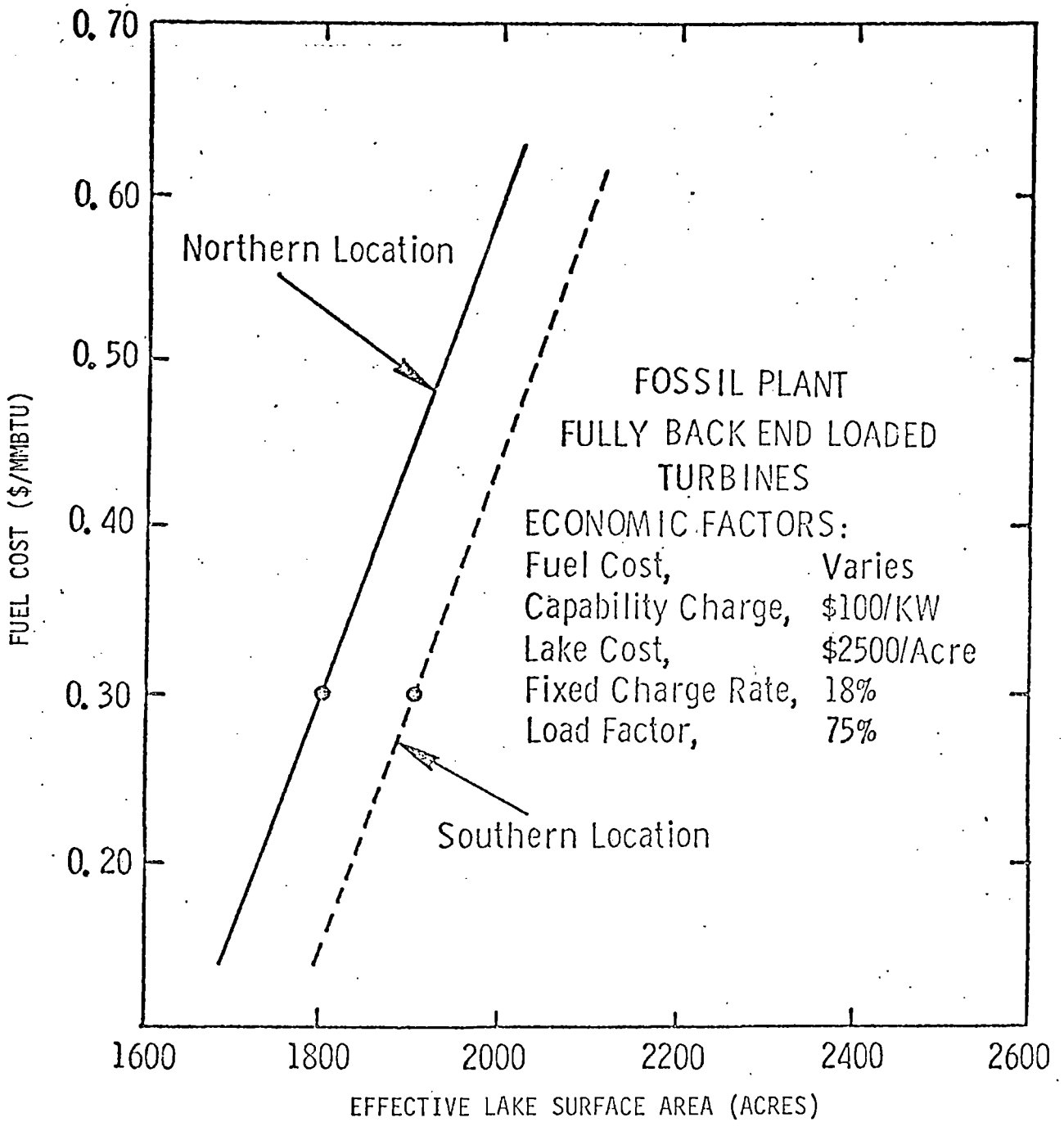


Figure 4-4. Fuel cost versus lake surface area

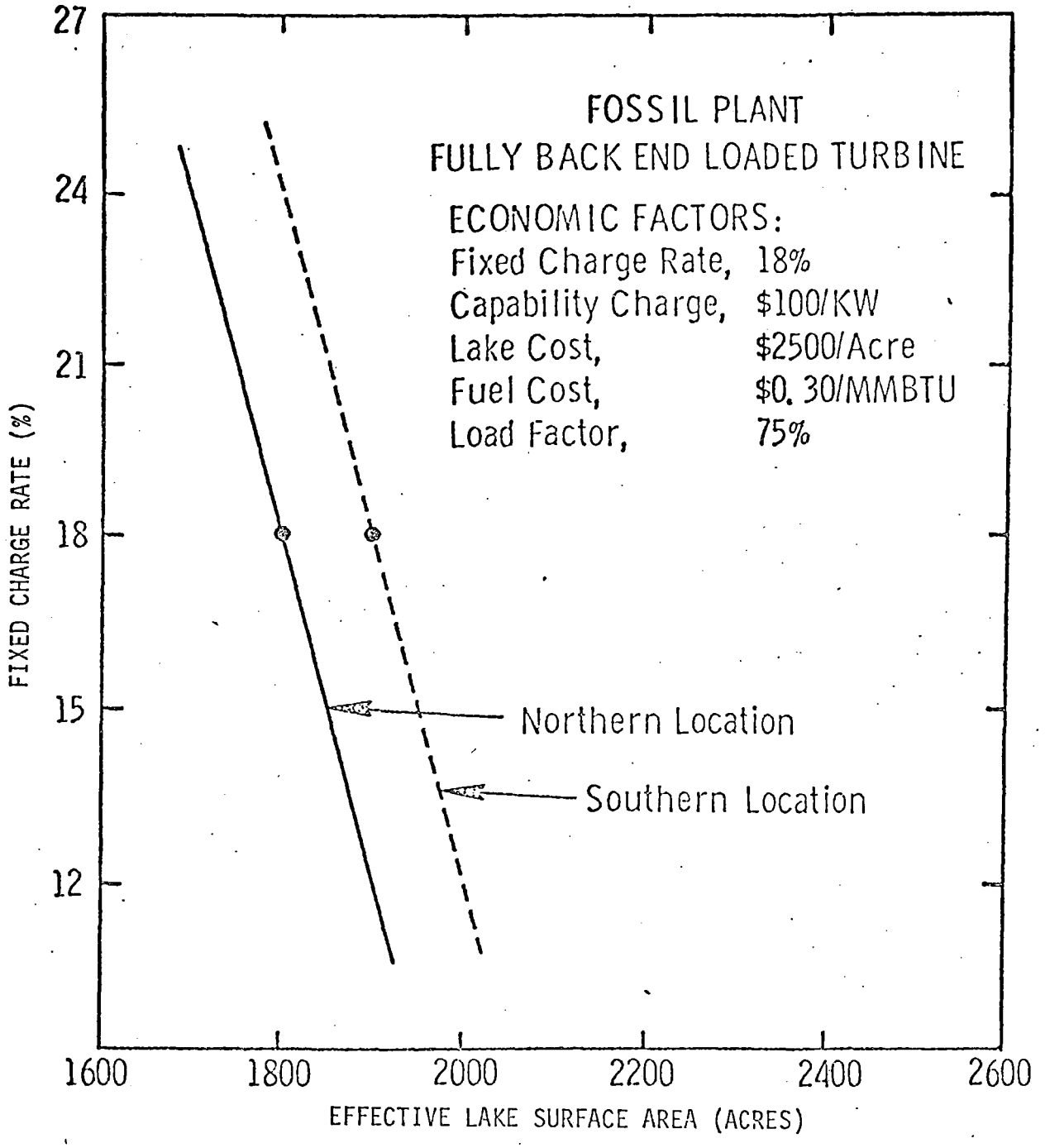


Figure 4-5. Fixed charge rate versus lake surface area

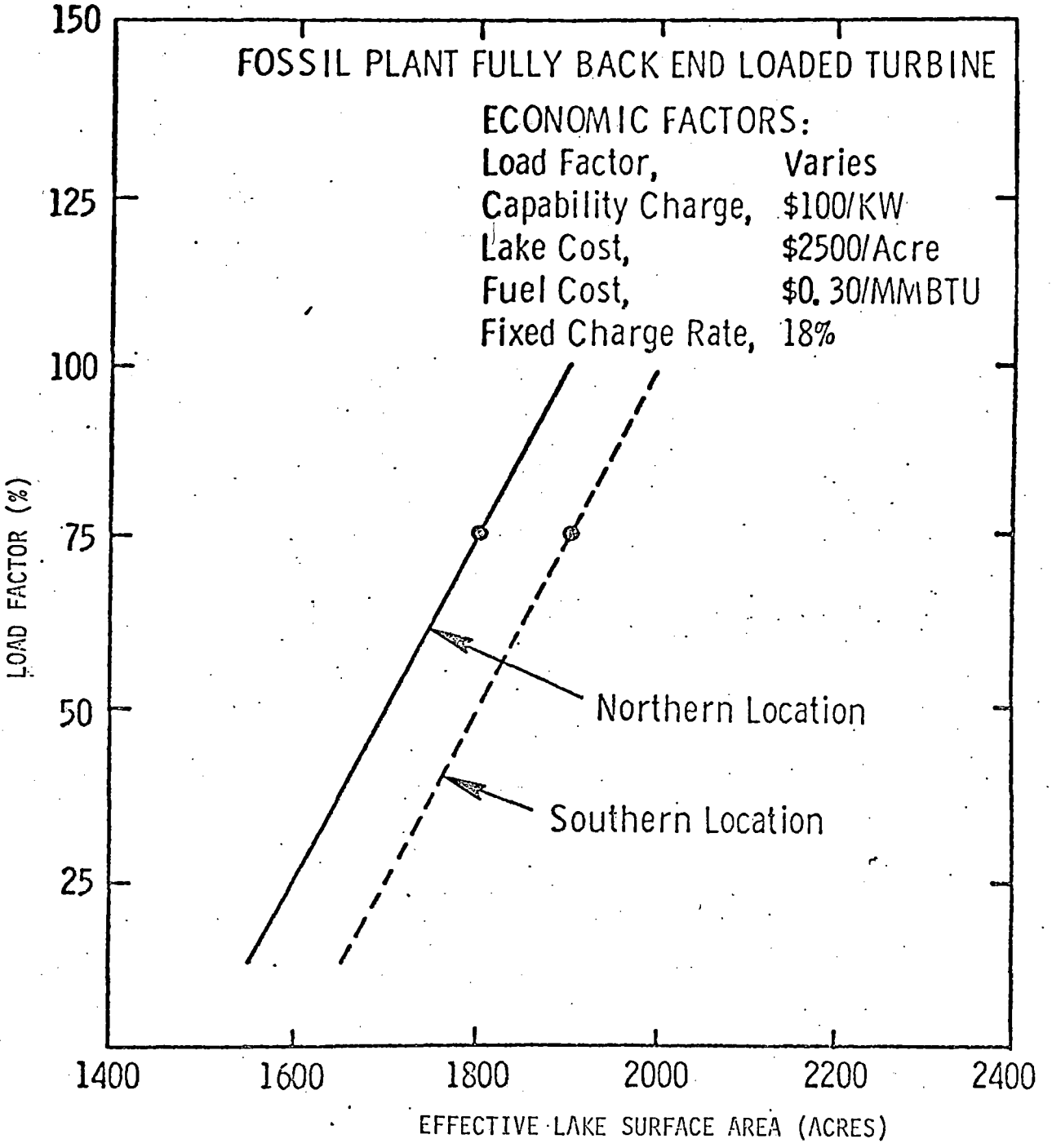


Figure 4-6. Load factor versus lake surface area

in fuel cost, fixed charge rate and load factor do not have as pronounced an effect on the optimum lake size for the ranges under consideration.

The optimum cooling pond size are not only affected by engineering and economic parameters, but also affected by Federal and State pollution control and regulation policy. In most design cases, these regulatory factors have become predominating. More discussion will be made in the last section.

4-3. Possible environmental impacts.

Potential environmental impacts due to cooling pond operation must be considered. Since heat transfer mechanism of cooling pond is similar to that of evaporative cooling tower, the impacts on environment must be similar. That is, vapor plume, fogging and water drift are the problems of concern. However, the cooling pond covers a large surface area and hence, these problems are generally less severe.

The section is intended to provide a brief summary of current work in this area. Huff and Vogal [30] studied cooling lakes in Illinois. They found that naturally occurring fogs normally occur when the dewpoint depression is 1.1°C or less. To have a dense fog, the depression is 0.6°C or less. In terms of saturation deficits, dense fog normally occurs when the deficit is 0.5 gram/kilogram or less.

There were several studies on spray cooling which may have direct application to the cooling lakes. Hoult [28] modelled a plume from a spray canal and predicted the length of visible fog. He concluded that "for typical spray pond installations, temperature well below 10°F are required for the fog to extend more than a few hundred feet downwind of the canal".

Roffman [50] carried out a series of theoretical calculation on drift deposition rates. He used a drift rate of 0.1% and found that the maximum drift deposition occurred at 10 m downwind of a spray canal (1.41×10^{-3} kg/m^2/day) under slightly unstable conditions. All information provided in the study point out the fact that drift is not a serious problem for spray cooling beyond 100 m and is practically non-existent beyond 200 m.

Stewart and Nelson [56] did two years research on spray cooling. Studies on the local meteorological effects (New York State area) of drift, icing and fogging were conducted. The pertinent results are quoted as follows:

"1. Drift rarely travels beyond 200 m from the spray cooler, and most of the drift is eliminated in the first 50 m.

2. Icing caused by drift does not exceed the distance traveled by the drift. Icing due to deposition by fog may occur at greater distance than icing due to drift. However, this is a rime icing which is not as dense and will melt more quickly.

3. Scavenging of droplets by screens, trees, and fences are effective in varying degrees. Conifers provide an aesthetically pleasing effect as well as an effective scavenger of both drift and fog.

4. In comparison to natural drift, such as that created at Niagara Falls, the spray cooling drift deposition is less than 1% at comparative distances."

Murray and Trettel [42] studied the steam fogging at the Dresden Nuclear Power Station in Illinois. At a site approximately 200 m southeast of the Dresden cooling lake, no increase over climatological expectation in fog-restricted visibility (less than 150 m) was detected. Out of some 400 observation of steam fogging, only five exceeded two hundred meters in horizontal extent. At Dresden the maximum distance at which rime ice was reported was 130 meters and this occurred twice. Six times riming occurred at a distance of 30 m or more.

In summary, the environmental impacts of cooling pond operation are expected to be small. Generally, it would occur in the periods during which the winds are light (2 m/sec or less), relative humidity is high (90% to 100%) and the ambient air temperature is below freezing.

4-4. A Case Study

In 1977 the writer was invited by a consulting engineering firm in Boston to do a feasibility study of using a cooling pond for one power station in the state of Arkansas. The section presents a sum-

mary of this study with a purpose that the interrationship between the engineering design and pollution parameters can be illustrated. The pollution parameters under consideration are the water temperature and distribution patterns.

The following plant site data were provided and used throughout the study:

power generation rate	2 x 800 MW
condenser heat load	2 x 4080 x 10^6 Btu/hr
condenser temperature rise	28°F
available cooling pond size	1190 acres

Weather data are presented as below.

	Design	Summer	Spring-Fall	Winter
Wind speed (mph)	7.3	7.3	8.3	9.0
Dry-Bulb Temp. (F)	97	80	65	48
Wet-Bulb Temp. (F)	78	71	56	41
Solar Radiation (Langley)	559	544	391	211

Other site information used for this study include:

* physical pond surface area.
* effective surface area. (i.e. excluding "dead-water" zones).
* average depth or depth distribution.
* maximum pond depth.

* pond shape. (width x length).
* pond orientation with respect to the prevailing wind direction.
* temperature of airflow and make-up water.
* temperature of seeping water and outflow.
* expected water temperature distribution in the vertical direction.
* pond inlet and outlet geometry and design information.

The major environmental constraints were in the area of reservoir water temperatures required for a 316a demonstation (EPA regulation). Briefly, there were three sets of critical thermal condtions to be satisfied. They were summarized as follows.

1. In March, April, and May, the cooling pond shall not have more than 50% of surface area at 84°F or above.

2. In August, the cooling pond shall not have more than 50% of surface area at 90°F or above.

3. The lethal temperature limit is at 96°F and the pond temperature must be less than 96°F over significant areas of the reservoir. (probably better than 2/3).

In this feasibility study another major concern is an economic impact on the new plant generation cost. Comparisons were to be made between the site with cooling pond and site with cooling towers. Because of these reasons, economic evaluations were included in this case study.

To illustrate the interrationships between the engineering design and pollution parameters, some of numerical results obtained in this feasibility study are presented and shown in Table 4-1 to Table 4-6.

TABLE 4-1

Condenser Inlet & Outlet Temperature

Type of Condenser: Single Pressure One Pass

Effective pond size: 1190 acres

	Max. back pressure (inches Hg. Abs)				
	2.5	3.0	3.5	4.0	4.5
Design	101.3-105.3	100.5-108.9	99.7-111.7	99.0-114.0	98.4-116.3
Summer					
Load 100%	94.8- 98.5	94.0-102.0	93.5-104.8	93.0-107.1	92.7-109.0
75%	91.0- 94.1	90.5- 97.0	90.0- 99.3	89.6-101.2	89.3-103.1
50%	88.2- 90.5	87.8- 92.6	87.4- 94.3	87.0- 95.6	87.0- 97.2
Spring-Fall					
Load 100%	81.2- 84.9	80.4- 88.4	79.8- 91.1	79.2- 93.3	78.7- 95.5
75%	77.6- 80.7	77.0- 83.5	76.3- 85.6	75.8- 87.4	75.3- 89.1
50%	74.0- 76.3	73.5- 78.3	73.0- 79.9	72.8- 81.4	72.5- 82.7
Winter					
Load 100%	69.5- 73.2	68.8- 76.8	68.2- 79.5	67.7- 81.8	67.0- 83.8
75%	65.8- 68.9	65.0- 71.4	64.5- 73.5	64.0- 75.6	63.8- 77.6
50%	60.7- 63.0	60.4- 65.2	60.0- 66.9	59.7- 68.3	59.4- 69.6

TABLE 4-2

Effects of Pond Size on the Condenser Inlet and Outlet Temperatures

Type of Condenser: Single Pressure, One Pass

Max. back pressure: 4.5 inches Hg Abs.

	Effective Pond Size		
	1190 Acres	1428 Acres	1785 Acres
Design	98.4-116.3	95.0-114.8	92.5-113.6
Summer			
Load 100%	92.7-109.0	88.8-107.3	85.0-105.0
75%	89.3-103.1	87.0-102.1	83.0- 99.7
50%	87.0- 97.2	83.2- 94.4	81.2- 93.3
Spring-Fall			
Load 100%	78.7- 95.5	74.8- 93.3	72.0- 91.9
75%	75.3- 89.1	73.0- 88.1	69.8- 86.1
50%	72.5- 82.7	69.5- 80.7	67.4- 79.5
Winter			
Load 100%	67.0- 83.8	63.4- 81.8	60.2- 80.1
75%	63.8- 77.6	60.4- 75.5	57.0- 73.3
50%	59.4- 69.6	56.7- 67.9	53.5- 65.6

TABLE 4-3

Comparison of One-Pass and
Two-Passes Condenser Arrangements

Max. Back Pressure: 4.5 Inches Hg Abs.

Effective Pond Size: 1190 Acres

	One-Pass Condenser	Two-Passes Condenser
Design	98.4-116.3	96.7-123.6
Summer		
Load 100%	92.7-109.0	91.0-116.4
75%	89.3-103.1	87.8-108.6
50%	87.0- 97.2	87.0-102.4
Spring-Fall		
Load 100%	78.7- 95.5	77.0-102.4
75%	75.3- 89.1	74.0- 94.8
50%	72.5- 82.7	72.5- 87.9
Winter		
Load 100%	67.0- 83.8	65.5- 90.9
75%	63.8- 77.6	62.5- 83.3
50%	59.4- 69.6	59.4- 74.8

TABLE 4-4

Annual Comparative Costs for Various Maximum Back Pressure *

Type of Condenser: Single Pressure, One-Pass

Effective Pond Size: 1190 Acres

Items	Max. back pressure (inches Hg Abs.) 2.5	3.0	3.5	4.0	4.5
Condenser Cost	7.85	4.35	3.25	2.72	2.40
Circulating Pump & Piping System Cost	36.99	19.14	13.55	10.84	9.22
Main Steam Turbine Cost	32.30	32.30	32.10	31.98	31.77
Total Investment	77.16	55.78	48.90	45.54	43.40
Indirect & Overhead Cost	19.29	13.94	12.22	11.38	10.85
Total Estimated Comparable Initial Cost	96.44	69.73	61.13	56.92	54.25
ANNUAL COSTS:					
Fixed Charges	15.43	11.16	9.78	9.11	8.68
Operating Cost	0.52	0.28	0.21	0.17	0.14
Generating Capability Penalty	0.40	0.26	0.37	0.45	0.61
Fuel Cost	74.19	74.88	74.98	75.10	75.22
Total Comparative Annual Cost	91.19	86.58	85.34	84.83	84.65

* All costs are in terms of million dollars.

TABLE 4-5

Cost Comparisons between the one-pass and two-passes condenser arrangements *

Max. back pressure: 4.5" Hg Abs

Effective Pond Size: 1190 Acres

Items	One-Pass Condenser	Two-Passes Condenser
Condenser Cost	2,40	2.93
Circulating Pump & Piping System Cost	9.22	6.45
Main Steam Turbine Cost	31.77	31.77
Total Investment	43:40	41.16
Indirect & Overhead Cost	10.85	10.29
Total Estimated Comparable Inital Cost	54.25	51.45
ANNUAL COST:		
Fixed Charges	8.68	8.23
Operating Cost	0.14	0.15
Generating Capability Penalty	0.61	0.61
Fuel Cost	75.22	75.25
Total Comparative Annual Cost	84.65	84.24

* All costs are in terms of million dollars.

TABLE 4-6

Effects of Pond Size on the Annual Comparative Costs * †

Type of Condenser: Single Pressure, One Pass

Max. back pressure: 4.5 inches Hg Abs.

Items	Effective Pond Size (Acres) 1190	1428	1785
Condenser Cost	2.40	2.25	2.15
Circulating Pump & Piping System Cost	9.22	8.48	7.91
Main Steam Turbine Cost	31.77	31.77	31.77
Total Investment	43.40	42.51	41.83
Indirect & Overhead Cost	10.85	10.62	10.45
Total Estimated Comparable Initial Cost	54.25	53.14	52.28
ANNUAL COST:			
Fixed Charges	8.68	8.50	8.36
Operating Cost	0.14	0.14	0.13
Generating Capability	0.61	0.61	0.60
Fuel Cost	75.22	75.21	75.19
Total Comparative Annual Cost	84.65	84.46	84.28

* All costs are in terms of million dollars

† Pond cost is not included in these calculations.

Table 4-1 shows the condenser inlet and outlet temperatures (pond intake and discharge temperature) with various condenser sizes. In this table the condenser size is implicitly expressed in terms of the maximum back pressure. Table 4-2 indicates the effects of pond size on the condenser inlet and outlet temperatures. It is seen that the larger the cooling pond, the lower the condenser inlet and outlet temperatures would be. The condenser arrangement would also have an impact on the water temperature. In general, the condenser inlet termperature (or pond intake temperature) would be lower for two-passes condensers than for one-pass condensers. This is clearly shown in Table 4-3. However, the reverse trend is true in the condenser discharge temperature.

Different engineering design configurations result at different costs. For each configuration described above, economic evaluation was made. The evaluation included initial costs, operating cost, fuel cost and generating capability penalty. Attention was focused to the cost differentials rather than the absolute costs. Table 4-4, Table 4-5 and Table 4-6 summarized the economic evaluations for the system configurations under consideration. It should be mentioned that all these results were obtained with the computer model developed by the writer [37].

In the feasibility study economic evaluations were also made for the case of the power plant with cooling towers. It was found that the plant with cooling towers would cost much more than that with a cooling pond. In other words, from engineering economic viewpiont, the plant should be located at the site where a cooling pond can be constructed and operated.

No study is completed unless environmental constrains are taken into account. Using the three constrains mentioned above, a typical condenser - pond arrangement was reviewed and recalculated. The results were presented in Table 4-7. It is seen that the environmental restrictions set by the regulatory agencies can not be met when the cooling pond system operates under design conditions. Even under the average summer conditions, the cooling system still has troubles with regulations. It was these restrictions that the cooling pond system was not selected for that particular power station in Arkansas even though it represented an economical choice.

The study also revealed that as the pond size increase, the pond water temperature would decrease. However, under the site conditions the cooling pond could not be increased too much in size. Therefore, it was concluded that the new generating facilities under consideration could not be located at the cooling-pond site.

TABLE 4-7

Predicted Values of the Condenser-Pond Arrangement "A"

Pond Size: 1851 Acres

Condenser "A": Single pressure, two passes, temp. rise 27 F, flow rate 302 x 10^3 GPM, heat transfer surface area 393 x 10^3 ft^2

	Intake & Discharge Temp. (F)	Max. Affected Area (%)		
		A_{84}	A_{90}	A_{96}
Design	96.7 - 123.6		*	*
Summer Load				
100%	91.0 - 116.4		*	45.6*
75%	87.8 - 108.6		53.5*	30.9
50%	87.0 - 102.4		46.8	20.4
Spring-Fall Load				
100%	77.0 - 102.4	40		
75%	74.0 - 94.8	26		
50%	72.0 - 87.9	12		
Winter Load				
100%	65.5 - 90.9			
75%	62.5 - 83.5			
50%	59.4 - 74.8			

Notes:

* = the cooling system cannot meet the regulation

A_{84} = related to the first regulation

A_{90} = related to the second regulation

A_{96} = related to the third regulation

5. DRY-TYPE COOLING TOWERS

The concept of direct cooling with air is simple and not new to industries. In petro-chemical and chemical industries, air cooling has been widely used for more than two decades. Other example are aircraft engine oil coolers, radiators in automobiles, air preheaters in steam generators, and regenerators in gas turbine systems. However, the use of air-cooled condensers or air cooled heat exchangers for removing the waste heat from power plants is still in an infant stage. Table 5-1 shows a list of power plants with dry-type cooling towers. In this country only two small units are in operation. One 330 mw power plant with a dry-type cooling tower is under construction.

5-1. Dry-type cooling tower fundamentals

The dry cooling systems may be classified on the basis of the configuration of the fluid flow path through the air-cooled heat exchangers. The three most common types of flow path configuration are:

1) Parallel Flow
2) Counter Flow
3) Cross Flow

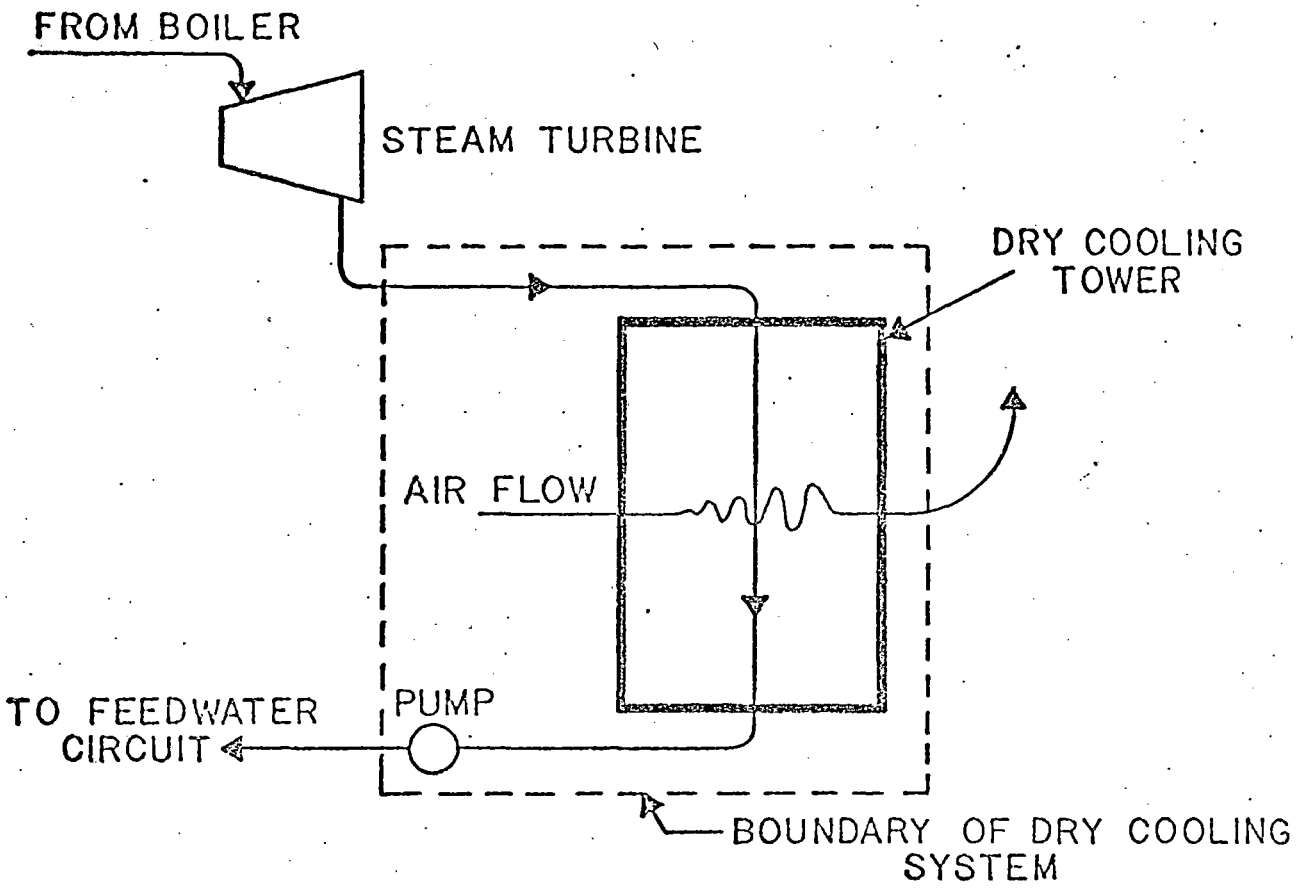
The most important difference between these three types lies in the relative amounts of heat transfer surface area required for a given heat load under the same conditions. The dry cooling systems can also be classified according to the types of condensers used. Figure (5.1) illustrates three types of dry cooling systems. They are:

1) direct system
2) indirect system with jet condensers
3) indirect system with tube condensers

TABLE 5-1

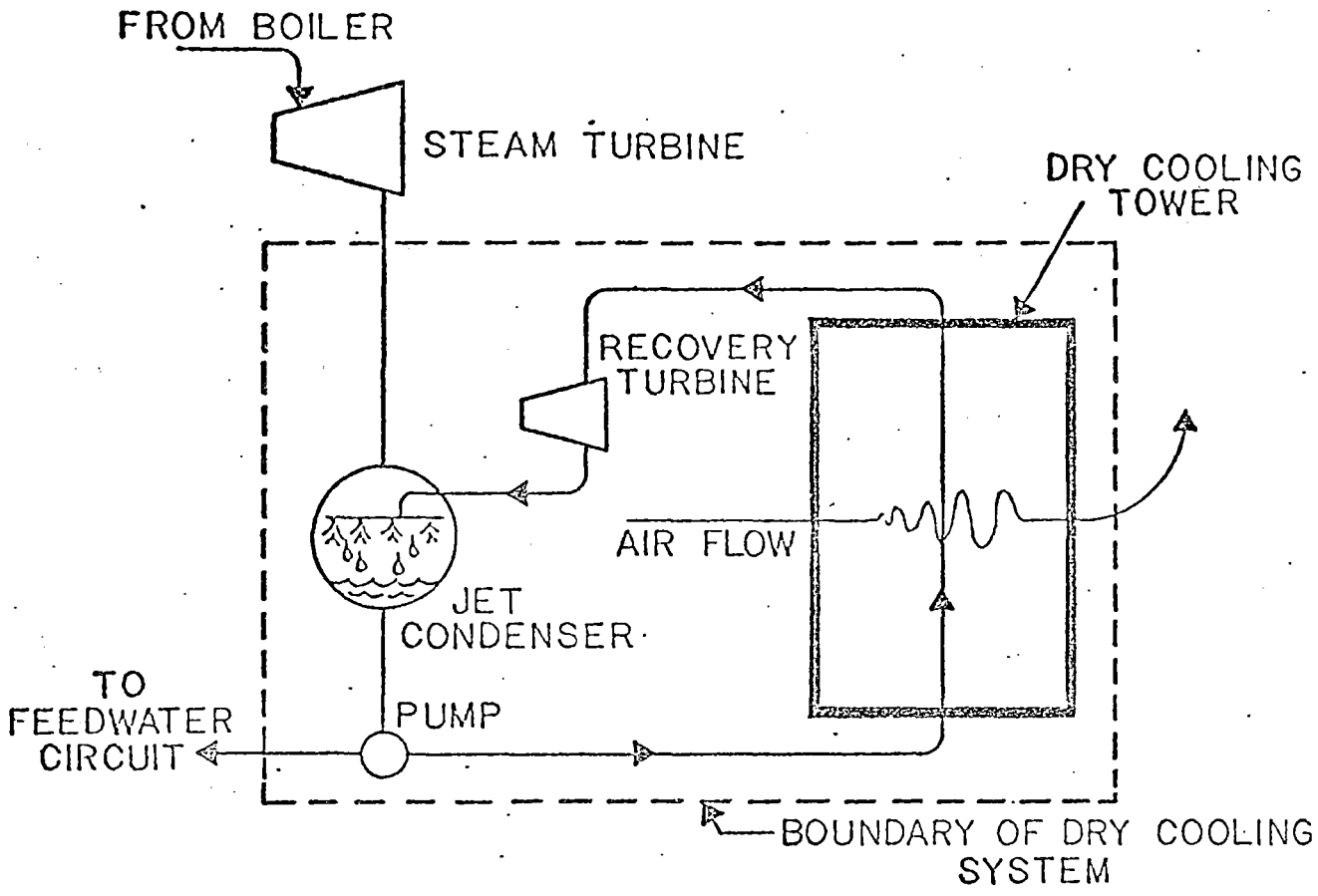
Generating Plants with Dry-Type Cooling Towers [17]

Location	Rating	Type of Dry Tower	Year Commissioned
Rugeley, England	120 mw	Heller	1962
Ibbenbüren, Germany	150 mw	Heller	1967
Wolfsburg, Germany	3-50 mw	GEA Direct	1961-67
Grootvlei, South Africa	200 mw	MAN/Birwilco (indirect)	1971
Gyöngyös, Hungary	2-100 mw 2-200 mw	Heller Heller	1969 Under Constr.
Razdan, USSR	3-200 mw	Heller	1970-72
Wyodak, Wyoming, USA	22 mw 3 mw	GEA Direct Direct	1969 1962
Utrillas, Spain	160 mw	GEA Direct	1970
Quetta, West Pakistan	7.5 mw	Baldwin-Lima-Hamilton (Direct)	1964
Bavaria	40 mw	GEA Direct	1960
Windhok, South Africa	3-30 mw	GEA Direct	1971
Switzerland	4.3 mw	GEA Direct	1969
Luxemburg	13 mw	GEA Direct	1956
Rome, Italy	2-30 mw	GEA Direct	1957
Cologne, Germany	28 mw	GEA Direct	1958
Sindelfingen, Germany	11 & 15 mw	GEA Direct	1960-61
Worms, Germany	5 mw	GEA Direct	1962
Chile	3.6 mw	GEA Direct	1963
Ludwigshafen, Germany	38 mw	GEA Direct	1966
Eilenburg, Germany	5.3 mw	Heller	NA
Dunaujvarus, Hungary	16 mw	Heller	1961



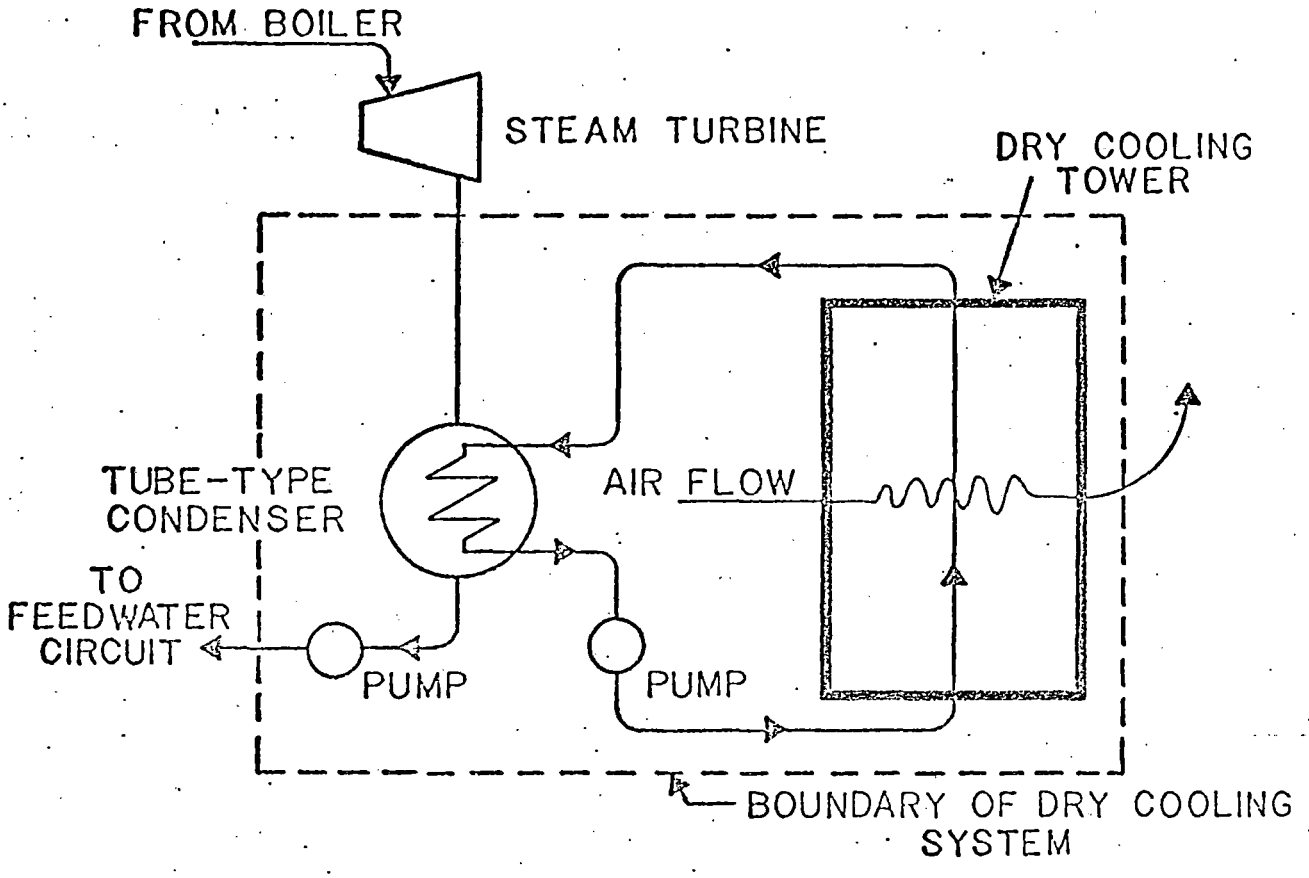
a/ DIRECT SYSTEM

Figure 5-1. Types of dry cooling systems (part a)



b/ INDIRECT SYSTEM WITH JET CONDENSERS

Figure 5-1. Types of dry cooling systems (part b)



c/ INDIRECT SYSTEM
WITH
TUBE TYPE CONDENSERS

Figure 5-1. Types of dry cooling systems (part c)

In industry, this direct system is often referred to as GEA (Gesellschaft Für Luftkondensation) system and the indirect system with jet condensers is named as Heller's system after the inventor, Professor L. Heller. As shown in Fig. (5-1), these dry cooling systems are similar to each other except in the area of condensation. In the direct system the air-cooled heat exchanger also acts as a condenser. The exhausted steam condenses within the heat exchangers and then returns to the boiler feedwater circuit. Obviously, the pressure in the direct system is always below atmospheric pressure. Air leakage would become serious, therefore it must be carefully considered both in design and in construction. In the indirect system there are no vacuum conditions and thus, the problem of air leakage does not exist. Compared with the direct system, indirect systems are more complex because of more pump facilities and condensers.

In all these dry cooling systems, the hot fluid (either circulating water or the condensing steam) flows inside the finned tubes while it transfers the heat to moving air outside by conduction and convection. In contrast to the wet cooling towers, there is no evaporation in the dry cooling towers and thus, it needs practically no water supply. The air motion in the air-cooled heat exchangers (dry cooling towers) is induced either by mechanical-draft fans or by natural-draft towers.

The performance equation for a dry-type cooling tower was derived in previous work [38]. It con be shown as

$$ITD = aQ^b \tag{5-1}$$

where: ITD = Initial temperature difference °F.

Q = Heat rejection, million BTU per hour.

a = Tower constant

b = Constant depending on natural or mechanical draft towers.

In the analytical work it was found that the tower constant (a) is related to the exchanger effectiveness and the minimum fluid capacity rate. The exponent b is equal to unity.

Equation (5-1) indicates a linear relationship between the heat load of dry cooling towers and initial temperature difference. This relationship has been confirmed by the performance of existing dry cooling towers in power plants. For example, the Ibbenburen and Rugeley Plants, which have been built and are in operation, have respectively [17]

$$ITD = 0.501\ Q^{0.717} \tag{5-2}$$

and

$$ITD = 0.247\ Q^{0.793} \tag{5-3}$$

As seen in these equations, the actual exponents are 0.717 and 0.793. They are little different from the analytical value. The difference may be due to the assumption used in the derivation of the performance equation. These assumptions are uniform fluid velocities, constant and uniform overall heat transfer coefficient and uniform temperature both at the inlet and outlet of heat exchangers. For the mechanical-draft type cooling towers, such as the one in the Neil Simpson Plant at Wyodak, Wyoming, the actual value of the exponent is equal to 0.91 and is indeed quite close to the analytical value.

Based on the above information, we take $b = 0.75$ for natural-draft and 0.91 for mechanical-draft towers. For a heat rejection 4940×10^6 Btu/hr. at ITD of 59.5°F the following equations can be established

$$ITD = 0.101\ Q^{0.75} \tag{5-4}$$

for natural-draft tower, and

$$ITD = 0.026\ Q^{0.91} \tag{5-5}$$

for mechanical-draft tower.

The performance of dry tower for various ambient air temperature and heat rejection loads can be illustated by the performance curves in which the ambient air temperature is plotted against turbines back pressure and heat rection. Fig. (5-2) shows typical curves of dry tower performance prepared according to Eq. (5-4).

Like the dry type cooling tower, the steam turbine has its own operational characteristics. The turbine theory indicates that the net heat rate will increase as the turbine back pressure increases. In addition, the increase in back pressure will also result in a decrease in the amount of work accomplished by the turbine, assuming constant steam flow. Since the turbine with dry-type cooling tower has different operational characteristics from those with wet-type cooling tower. The following are prepared to serve as a reference.

Fig. (5-3) shows an estimated turbine heat rate variation with elevated back pressures. The curve 1 was prepared for the conventional steam turbine which can only operate with the back pressure at 5 inches

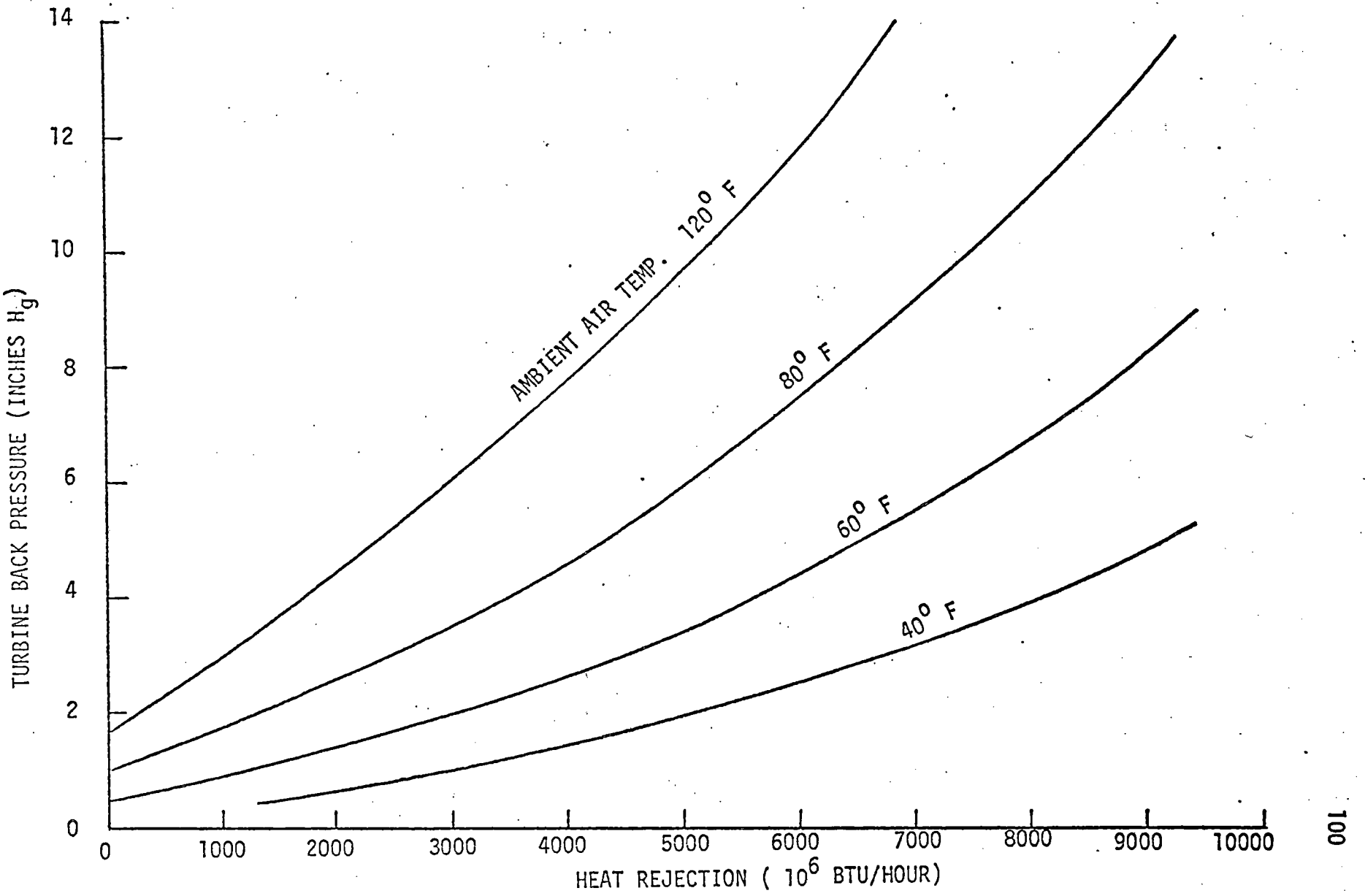


Figure 5-2. Dry tower performance curves

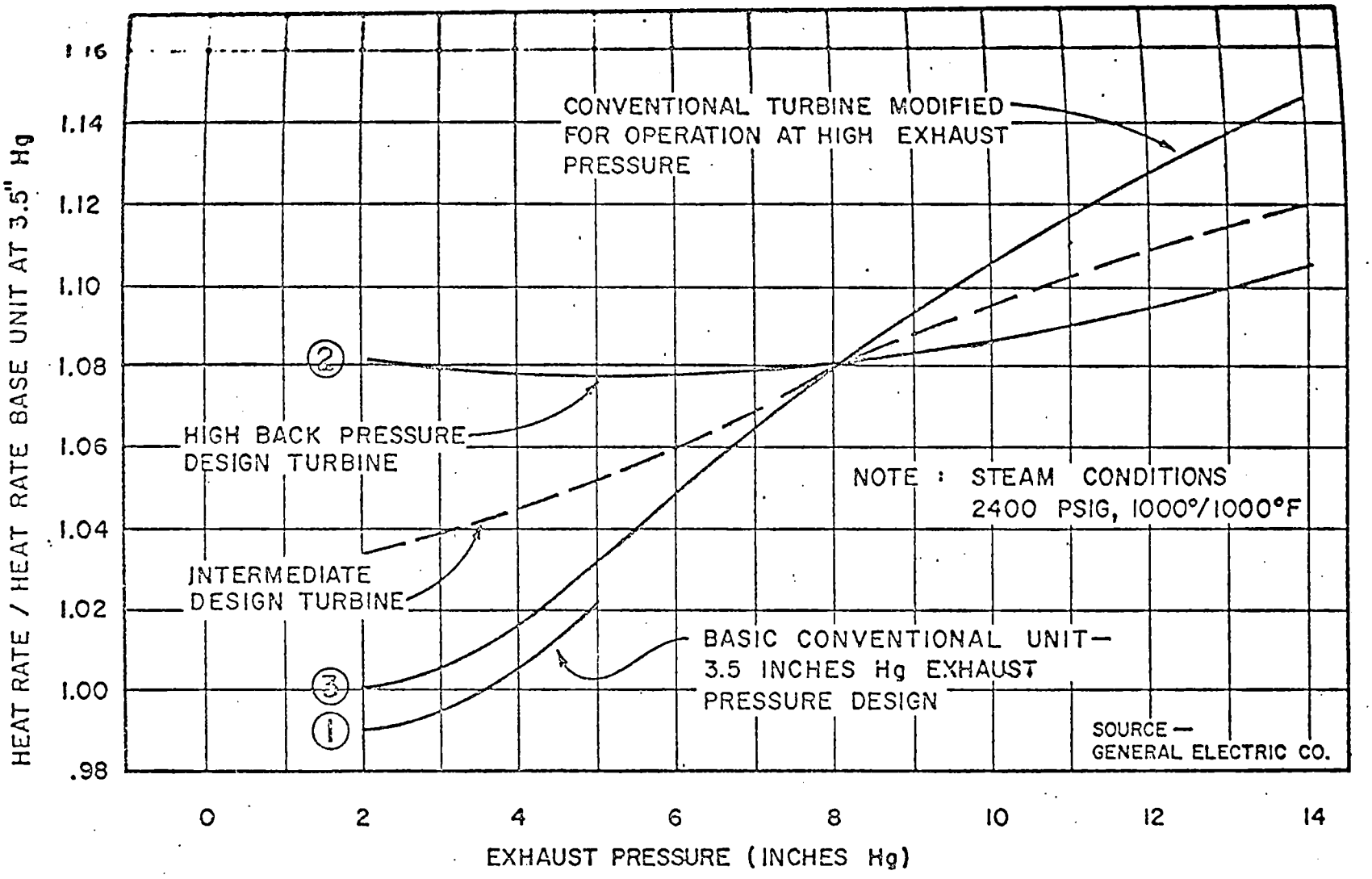


Figure 5-3. Estimated turbine-generator, full load, heat rate variation with elevated exhaust pressures

Hg. abs. or below. When the dry-type cooling tower is used, the turbine may have to operate at an elevated back pressure. Because of these reasons, present steam turbine must be modified or new turbine design would be required. The curves 2 and 3 were predicted for the new and modified steam turbines.

A change in the turbine net heat rate would result in a change in the generating capacity. The following equation can be used for the calcualtion

$$\Delta KW = \frac{-\Delta HR}{1+(\frac{\Delta HR}{100})} \qquad (5\text{-}6)$$

Where ΔKW = percentage change in kilowatts output

ΔHR = percentage change in turbine net heat rate

Fig. (5-4). shows the effect of increased turbine back pressure on total heat rejection to condenser at various turbine loads for an 1000 mw fossil-fueled plant. If the turbine heat rejection curves from Fig. (5-4) is plotted on Fig. (5-2), as shown in Fig. (5-5), the intersections of the turbine performance curves and the tower performance curves represent the turbine back pressure which will prevail with varying turbine-generator loads with ambient air temperature.

From the combined curves it is seen that the turbine output will drop off rapidly as the ambient air temperature increases. It is these drawbacks which prevent dry-type cooling towers from being accepted in electric power generation business.

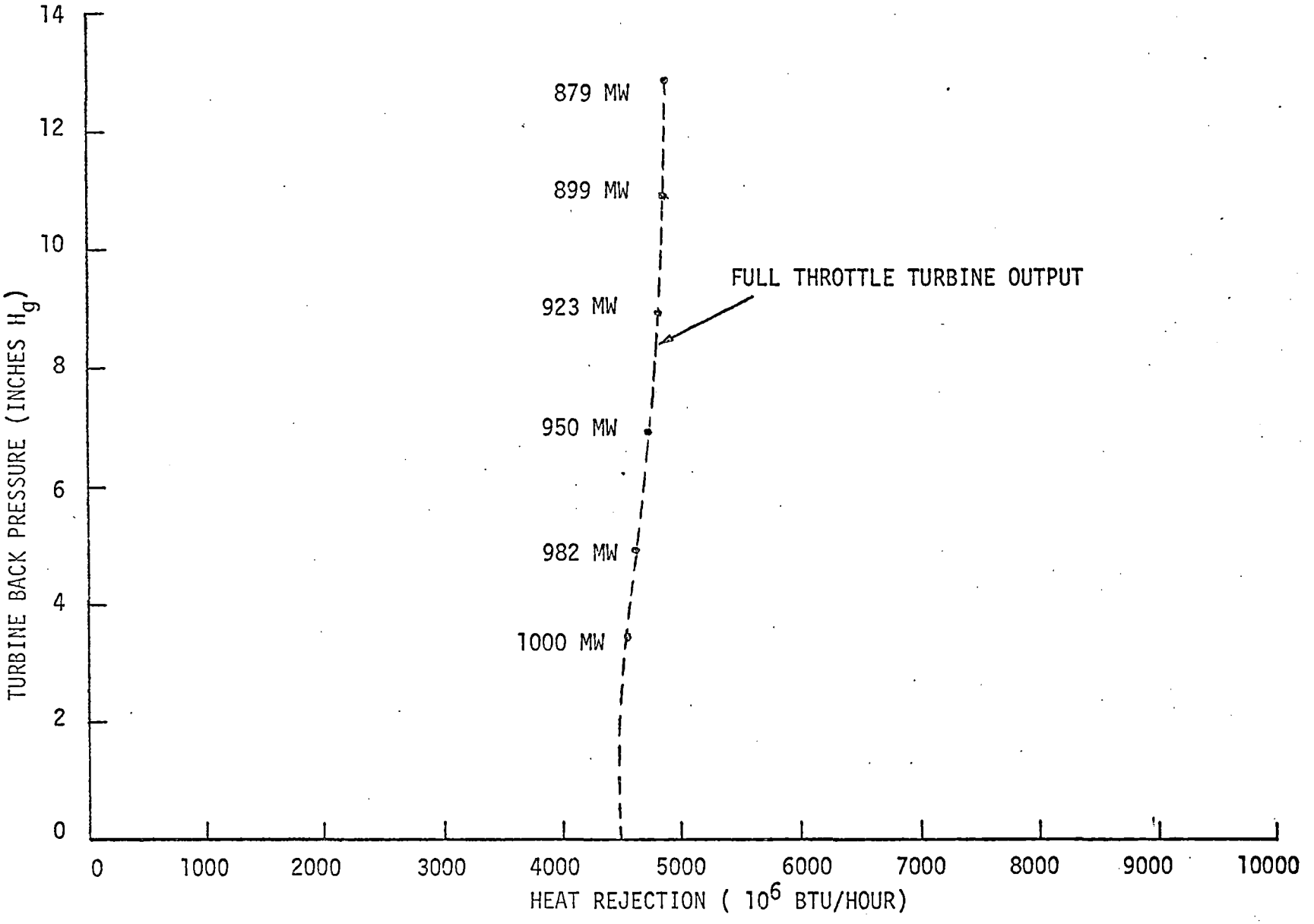


Figure 5-4. Steam turbine performance curve

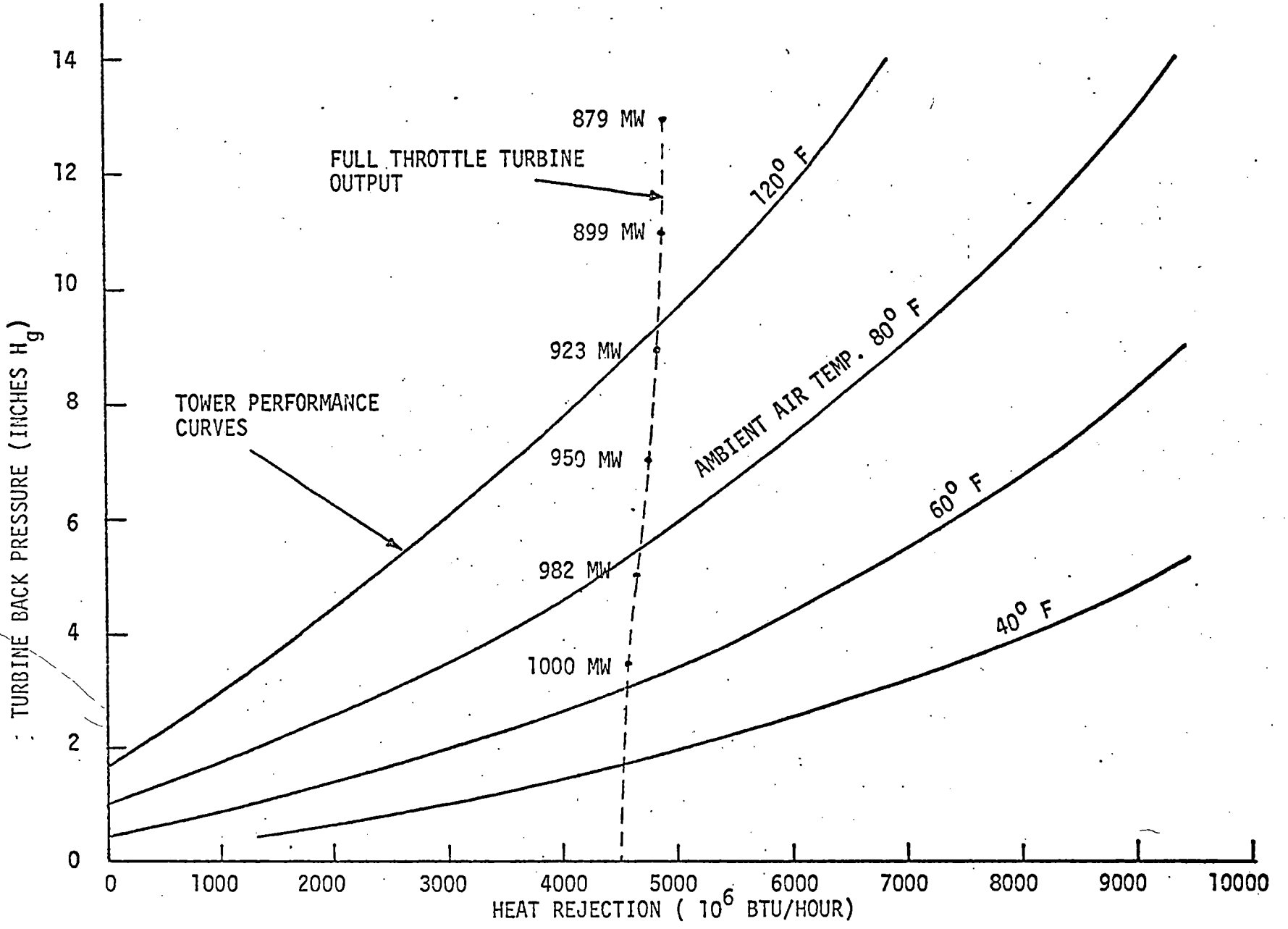


Figure 5-5. Turbine and cooling tower combined performance curves

5-2 Waste Heat Emission from a 1000 mw Plant

Waste heat emission rate is of interest to engineers as well as general public. These information is needed for prediction of possible environmemtal impacts including the tower plume behavior and characteristics. In this section a typical 1000 MW fossil-fuel plant is used for an illustration.

The turbine operates with steam conditions 35000 psig, and 1000°F. It has one reheater with maximum temperature 1000°F and eight feedwater heaters with final feedwater temperature 510°F. At the condenser pressure 3.5 inches Hg, the net turbine heat rate is around 7878 Btu/Kw.hr. and the nominal output is approximately equal to 1000 MW. The steam turbine is assumed so modified that it can operate at condenser pressures up to 15 inches Hg. abs. Table 5-2 summarizes the turbine net heat rate and output at various condenser pressure. Also included are the waste heat rejections.

In the hypothetical power plant the dry-type cooling tower is designed at the maximum head load 4940 x 10^6 Btu/Hr. The design initial temperature difference (ITD) is 60°F. When it is combined with the steam turbine system, the condenser pressure is seen to be affected by the ambient air temperature. Table 5-2 shows the relationship between the ambient air temperature and the condenser pressure and the turbine output. The hypothetical power plant will generate electric power at the rate of 1000 MW when the ambient air temperature is 64^{o}F. As the ambient air temperature increases, the turbine output drops rapidly. If the ambient air temperature around 99^{o}F. The turbine output is only 923 MW, and reduced by 7.7 percent.

Table 5-2
A typical 1000 MW fossil-fuel power plant performance.

Back Pressure. inches Hg. ghs.	3.5	5.0	7.0	9.0	11.0	13.0
Turbine Net Heat Rate Btu/Kuh	7996	8138	8390	8610	8799	8957
Turbine Net Output KW	1,000,000	982,000	950,000	923,000	899,000	879,000
Waste heat Rejection 10^6 BTU/Hr.	4583	4640	4728	4797	4842	4873
Ambient air temp. (F).	64.2	76.9	89.2	98.8	107.0	114.0

It is of interest to notice that the waste heat rejection increases while the turbine output decreases. At the ambient air temperature 64°F the rejection rate is approximately equal to 4583 MBTU/HR. But as the ambient air temperature reaches 99°F the waste heat rejection rate becomes 4797 MBTU/HR. and increased by 4.6 percent.

Waste heat from dry-type cooling tower is in form of hot air. It is rejected either from mechanical-draft or natural-draft towers. The exit velocity is around 17 feet per second for natural-draft tower and 26 feet per second for mechanical draft. Other comparisons are presented in Table 5-3. All these figures are approximate in nature and valid for one particular design. For example, if the design initial temperature difference is taken at 70°F instead of 60°F. The tower heat transfer surface will become smaller, resulting in a cost reduction. At the same time the turbine performance will deteriorate and result in a poor fuel economy. In practice, an engineering optimization is needed in selection of the intial temperature difference.

5-3 Plume Behavior & Characteristics

Plumes from dry-type cooling towers are different from those of wet cooling towers. The air receives the waste heat by convection and conduction in the dry-type tower and has its temperature increased. Because of no evaporation involved, the water vapor content remains constant and thus, visible plume does not exist.

Table 5-3
Comparisons of Mechanical-draft and Natural-draft day-type cooling towers for a typical 1000MW plant.

Hours of Comparison	Natural-draft towers	Mech.-draft towers
Exit air velocity, fps	17	26
Exit air temp. F	101	110
Exit air flow rate 10^6 lb/Hr.	516	415
Typical Tower Dimensions.	Height 713 ft. Top Diam. 440 ft. Bottom Diam. 510 ft.	Height 115 ft. length 833 ft. Width 354 ft.

Like the vapor plume, the dry-type cooling tower is a complicated process. Mathematical management of this dispersion process requires two major segments: 1. The effluents rise on their own momentum and buoyancy forces when atmospheric stability and turbulence are operative, but their influence on plume behavior is secondary and 2. The momentum and buoyancy forces become relatively weak and the effluent plume starts to dilute by mixing with surrounding ambient air. Fundamentally, the two parameters of atmospheric system which have a strong influence on the dispersion of hot gases are wind velocity and characteristics of turbulence (atmospheric stability).

Plume calculational procedures are similar to those of wet cooling towers. For simplicity, calculational results for a 1000 MW fossil-fuel plant are presented. These include the plume rise, penetration of inversion and entrainment of ambient air. [65]

In a stable atmosphere the plume rise decreases as the wind velocity increases. Fig. (5-6) indicates a general relationship between the wind velocity and plume rise. For a 1000 MW fossil-fuel plant, using a natural-draft tower, the plume rise calculation was done with assumption that the plant operates under design conditions. If the operation is off from the design conditions the following equation is used for prediction of plume rise:

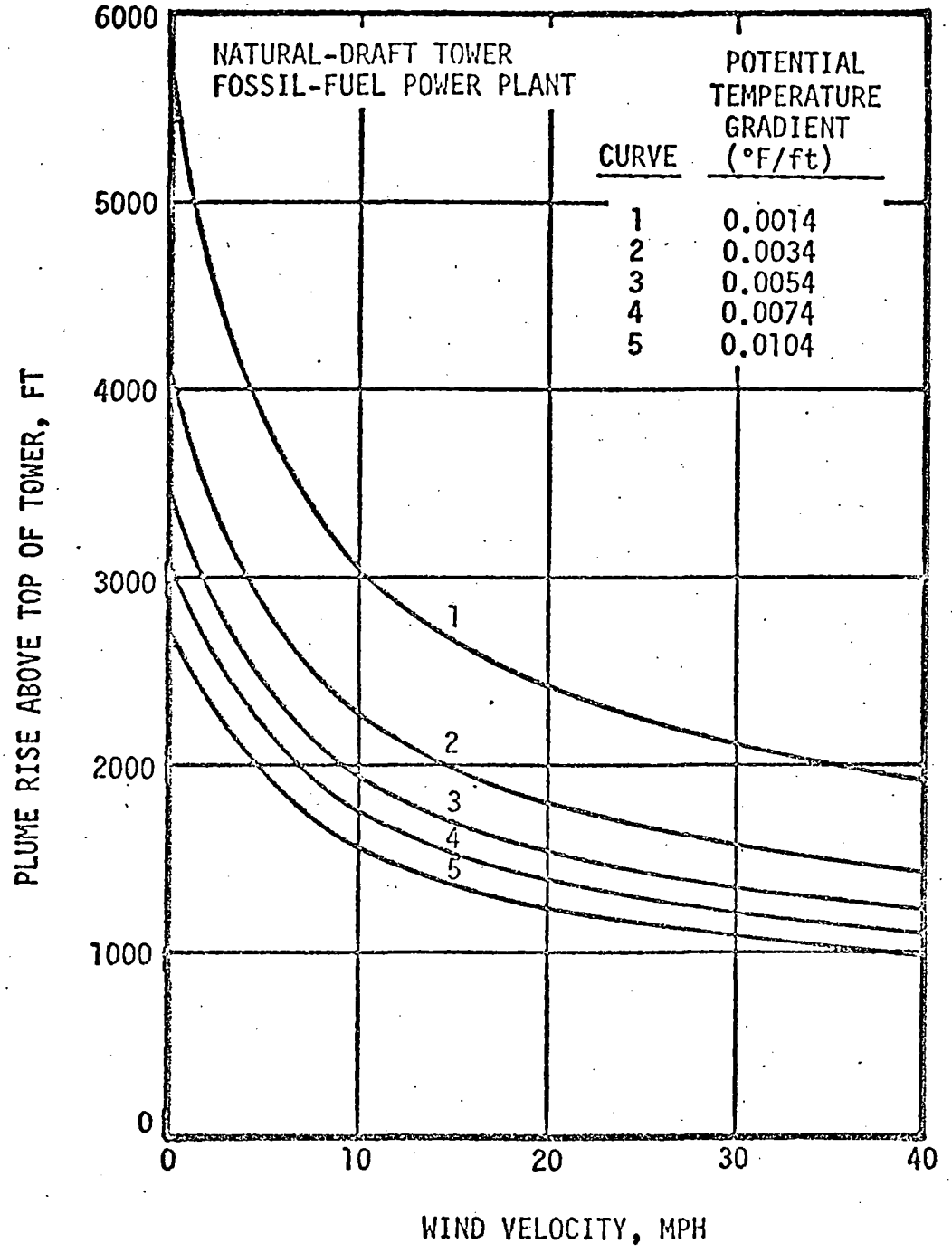


Figure 5-6. Plume rise above a natural-draft dry cooling tower. (stable conditions)

$$\frac{\Delta h}{\Delta h_{design}} = \left(\frac{\dot{Q}}{\dot{Q}_{design}} \right)^{1/3} \qquad (5\text{-}7)$$

where Δh = plume rise above the top of the tower

Q = heat load

Eq. (5-7) is also useful in predicting the plume rise for a multiple dry cooling towers, each supplying an equal fraction of the total plant heat rejection requirements. For example, when two towers are used, each rejecting 50% of the total waste heat, the plume rise above each tower is 79% of that for a single tower providing the total waste heat rejection for the plant.

Plume rise through a neutrally-stratified atmosphere is shown in Fig. (5-7). For comparison the plume rise in a stable stratification is included. It is seen that the plume rise in a neutrally-stratified atmosphere is larger than those in a stable system. Since the mathematical model available is not yet in a perfect level, the calculated plume rise may be different from the expected value. The 10% error band shown in Fig. (5-7) has been suggested for the actual plume rise in a flat, level terrain and the 40% error band for those in the presence of a significant terrain feature or the existance of a nearby water body.

The mathematical model used for predicting the plume rise through an unstable atmosphere is still not available at the present time.

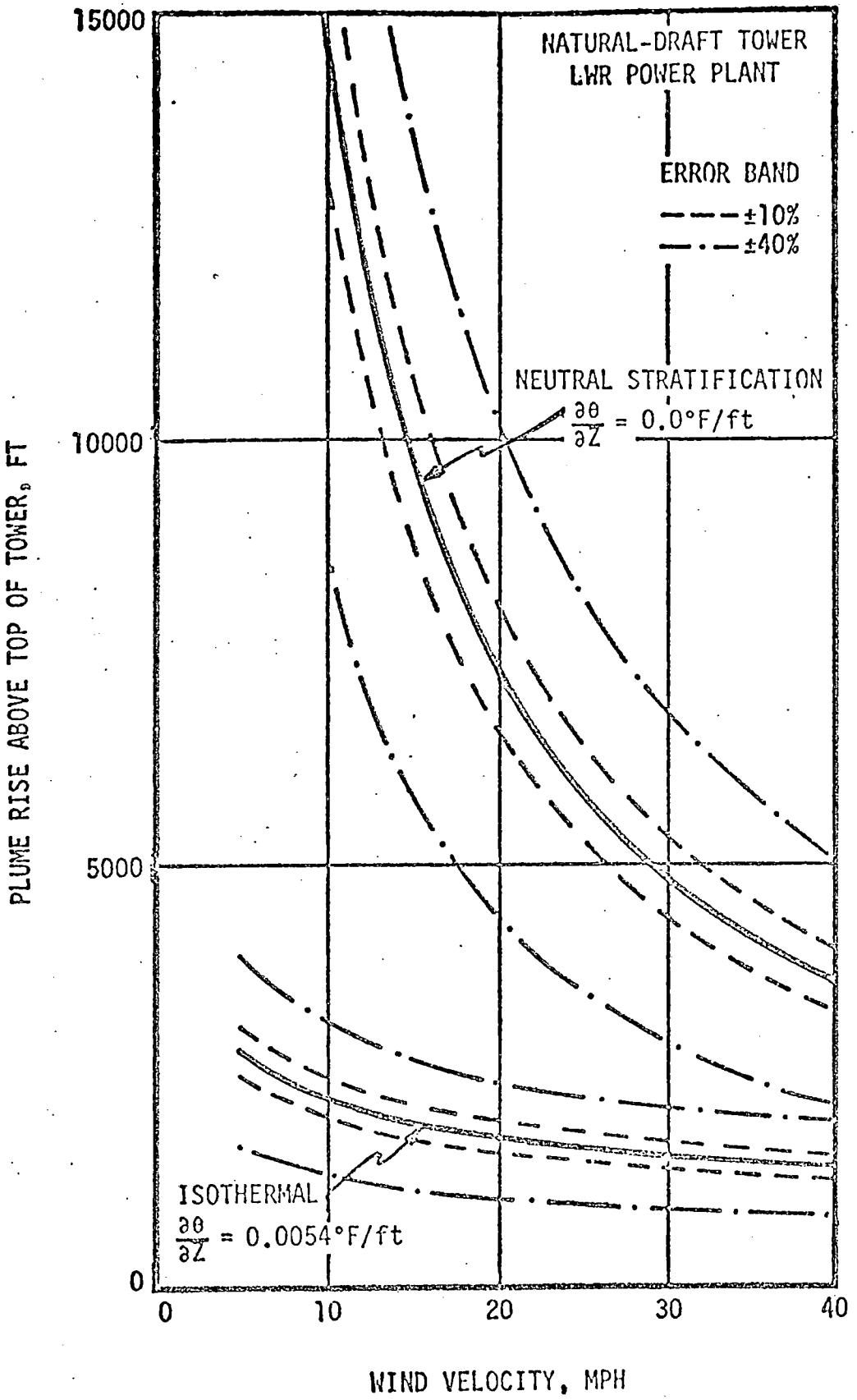


Figure 5-7. Plume rise above a natural-draft dry cooling tower. (neutral conditions)

Suggestion has been made that the plume rise through an unstable atmosphere is taken as equal to that through a neutral atmosphere.

Calculational procedures for plume rise of mechanical-draft cooling towers are similar to those for natural-draft towers. But the additional concept, (equivalent source area) is needed for an accurate prediction. The equivalent source area is an idealization of mechanical-draft tower exit as such that the plume rise calculated will be equal to that measured in the field. It seems that this term is used for adjustment of present mathematical model used for predicting the plume rise from mechanical-draft cooling towers.

Fig. (5-8) indicates the plume rise above the basic 12-fan unit of a mechanical-draft dry cooling system for an 1000 MW fossil-fuel plant. It is seen that the predicted rise heights are less than those from the natural-draft cooling tower. It is due to the fact that the heat rejection from the equivalent source area (12 units) is only 25% of the total waste heat.

Fig. (5-9) shows the calculated plume rises with three different equivalent source areas. The curve "1" presents the maximum value of plume rise. This situation will occur when the wind direction is parallel to the long axis of the cooling tower system. The high plume rise results from the max. entrainment of heated air from adjacent units. The min. plume rise presented by curve "3" are those calculated by omitting the entrainment effects. It is seen that the plume rise by 12 fan units equivalent source area is almost in the middle of these

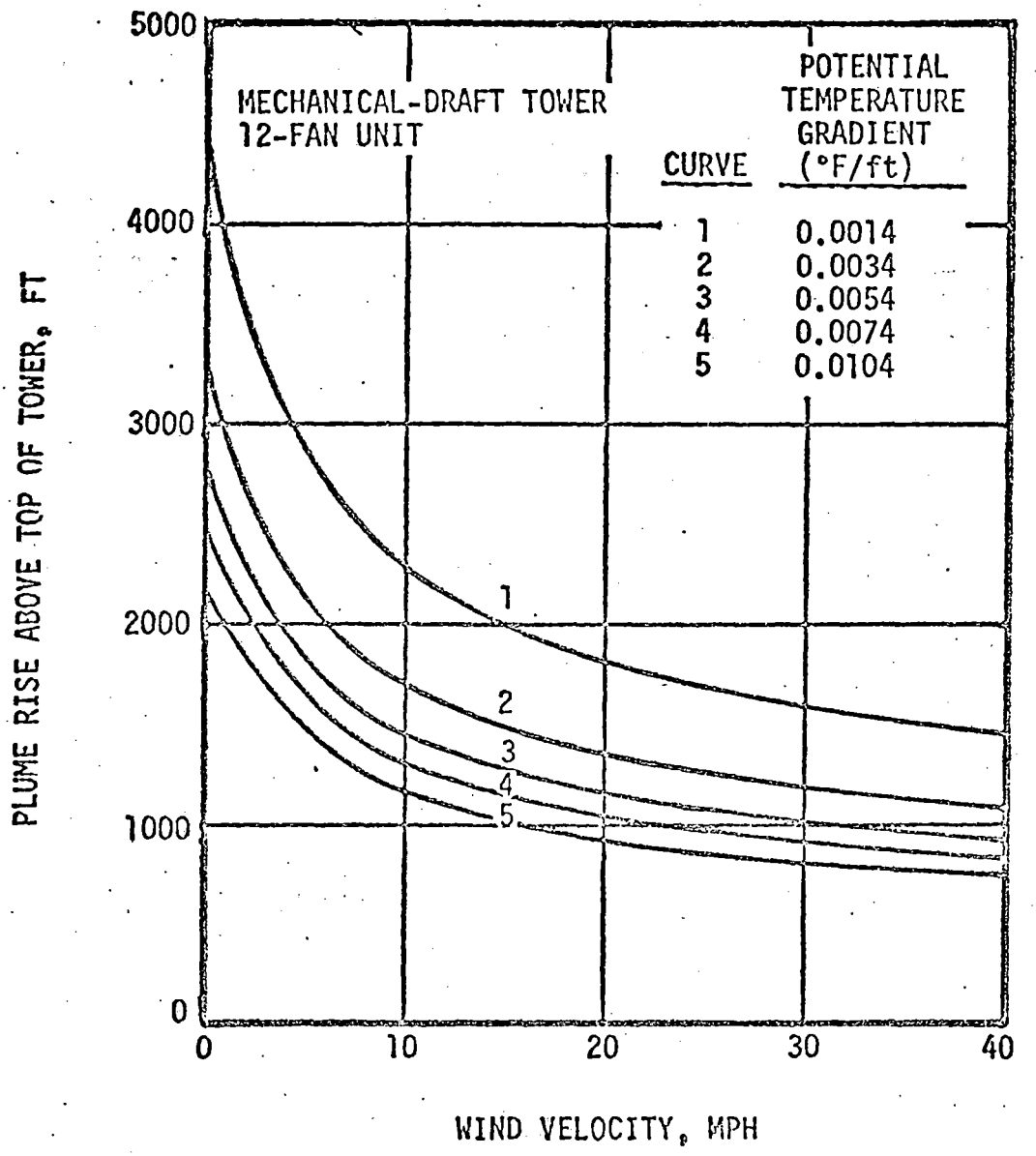


Figure 5-8. Plume rise above a basic 12-fan unit of mechanical-draft dry cooling tower

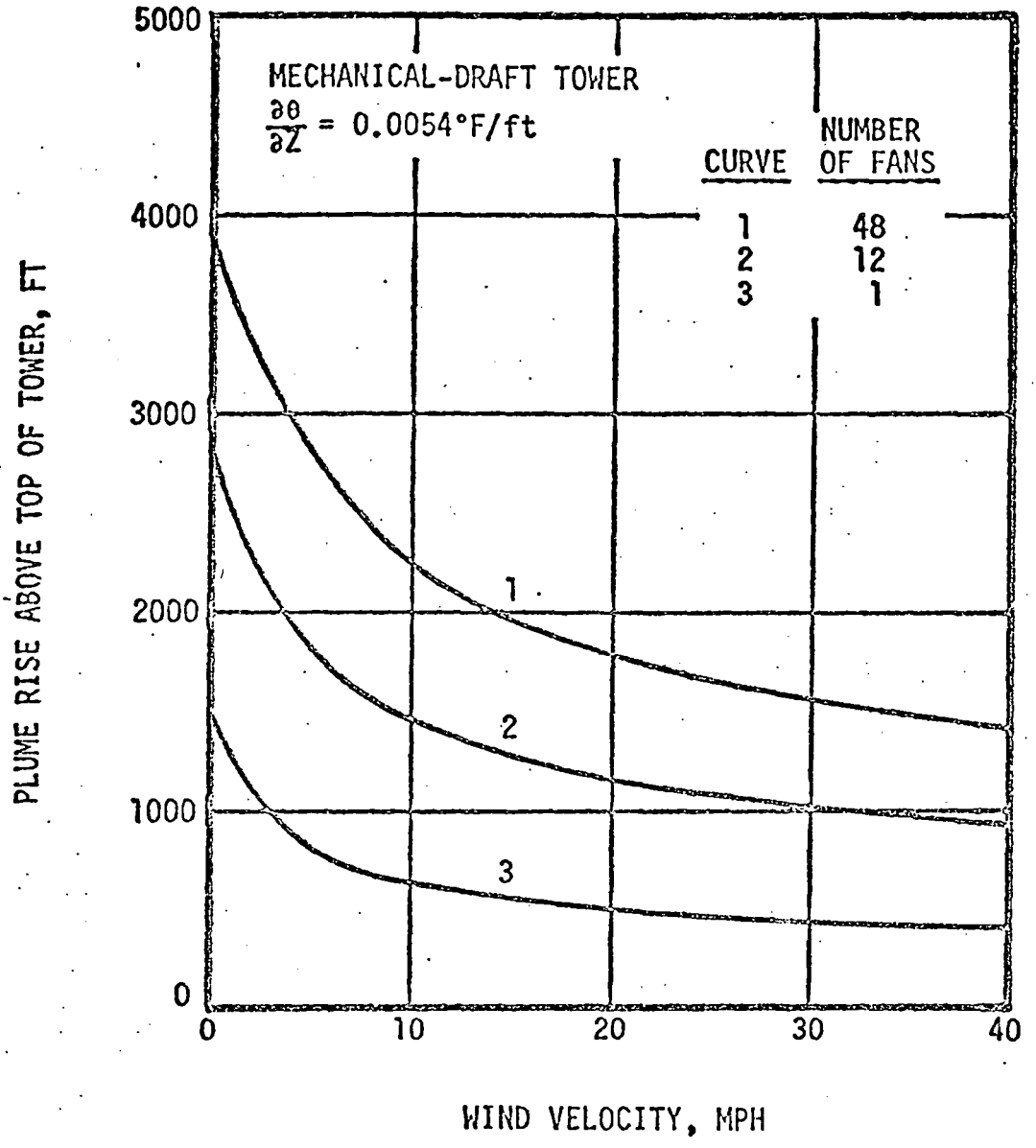


Figure 5-9. Effects of equivalent source area on the plume rise

two extremes. Field data indicates that when the wind direction is normal to the long axis of the cooling tower system, the12 fan units for equivalent source area would produce reasonably good calculations.

It is of interest to notice that while the dry-type cooling tower plume is invisible, it has a good ability to penetrate the inversion layers, which may exist in the atmospheric system. The height of penetrable ground-based inversions can be easily obtained from the plume rise calculations. In general, the plume can penetrate those inversions whose height above the top of the tower is less than the calculated plume rise. In addition to the ground-based inversions, there may also exist the elevated inversion layers in the atmospheric system. Fig. (5-10) shows the height above the top of the tower to the base of a penetrable elevated inversion layer in terms of wind speed and inversion strength. It is seen that as the strength of the inversion increases, the height above the top of the tower at which the inversion can exist and still be penetrated by the plume will decrease. For the case of mechanical-draft cooling towers, the results shown in Fig. (5-11) are based on the 12-fan equivalent source area.

As the effluents from dry-type cooling towers rise, they will entrain the ambient air through the interface between the plume and the surrounding air. The entrainment ratio is defined as the ratio of entrainment rate to the effluent rate. Fig. (5-12) indicate the entrainment ratio in term of wind speed and the elevational level. Like other presentations in this section, there calculated results are valid only for a 1000 MW fossil-fuel power plant.

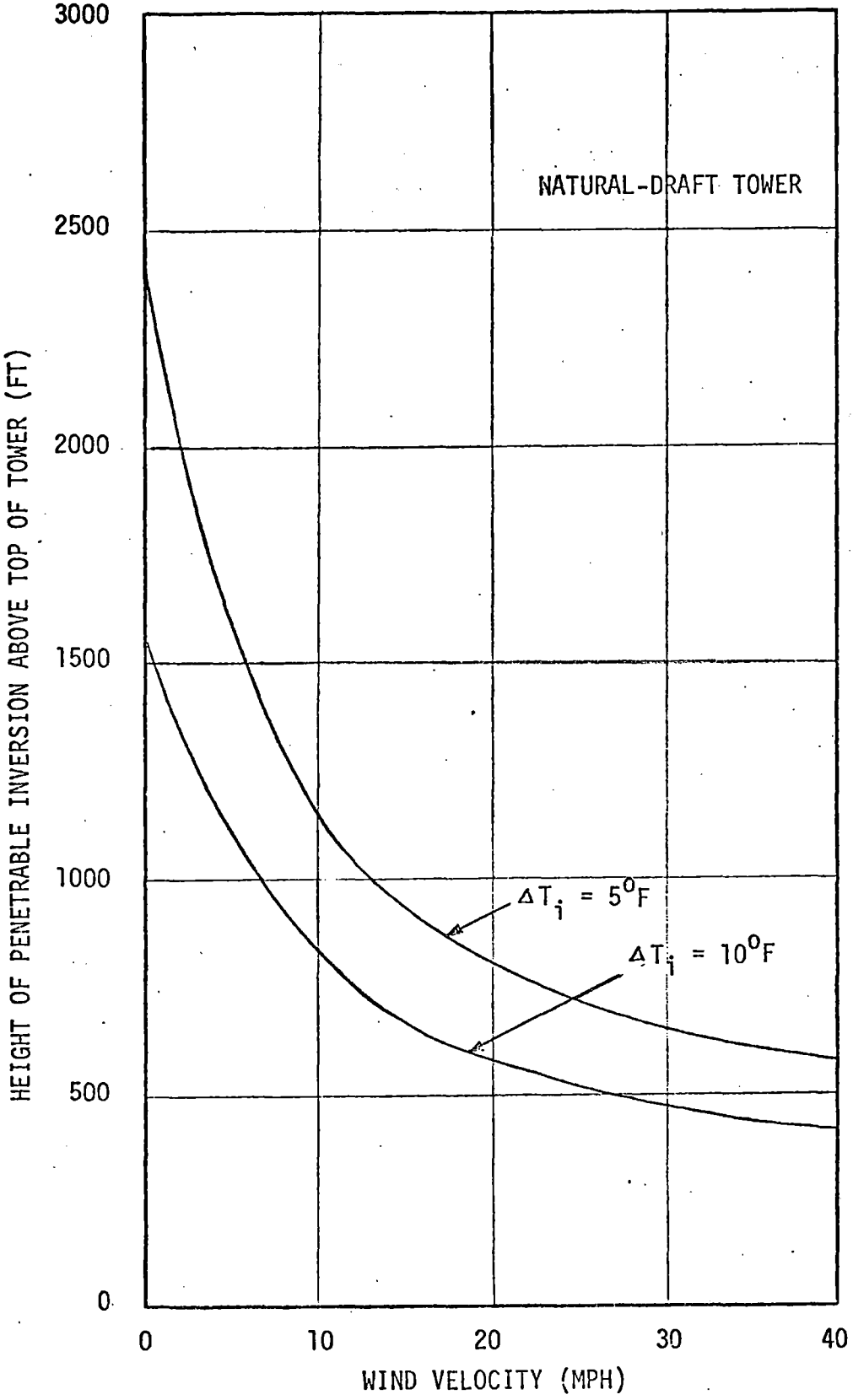


Figure 5-10. Height of Penetrable Inversion above top of natural-draft cooling tower

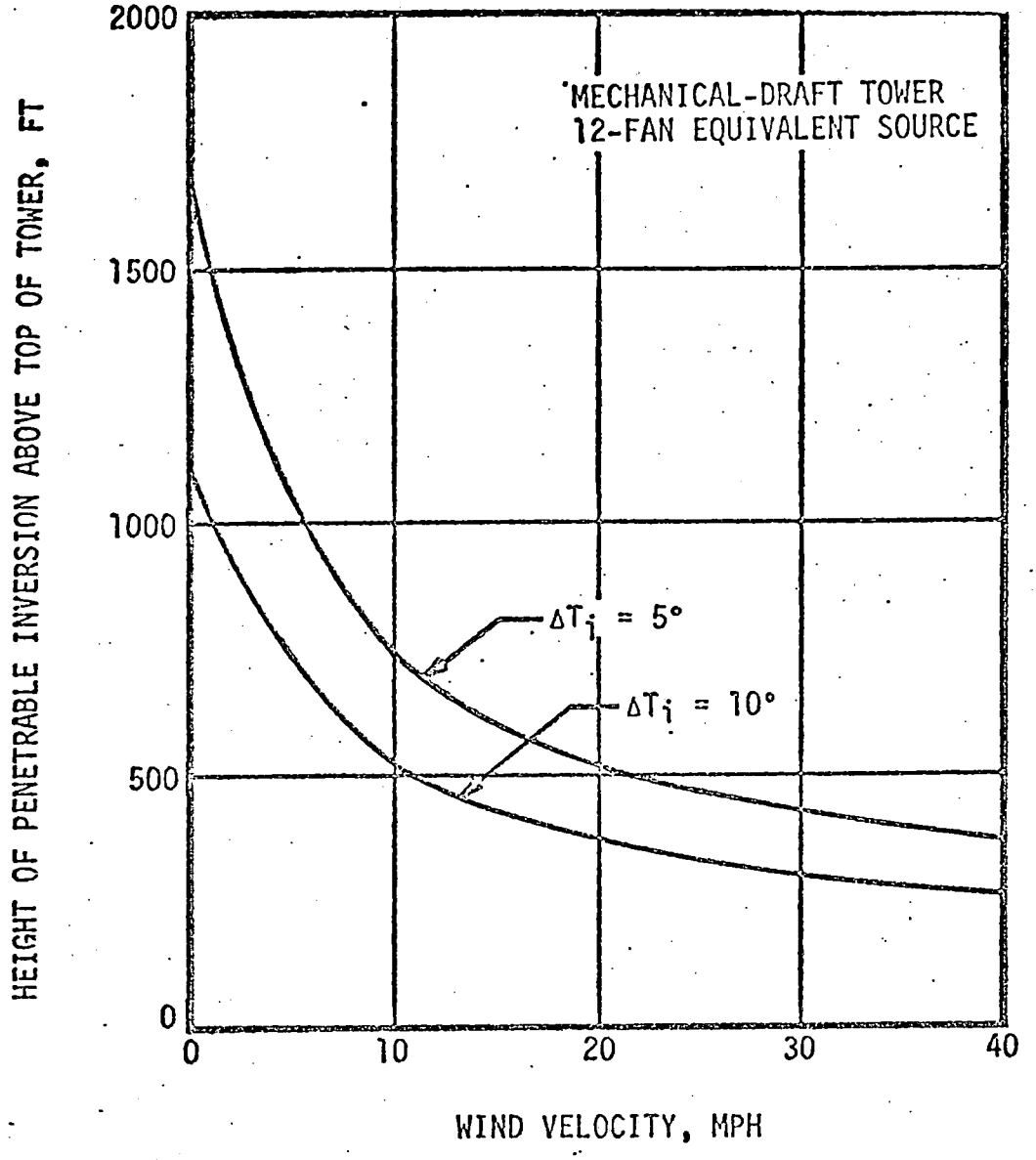


Figure 5-11. Height of Penetrable Inversion above top of mechanical-draft cooling tower

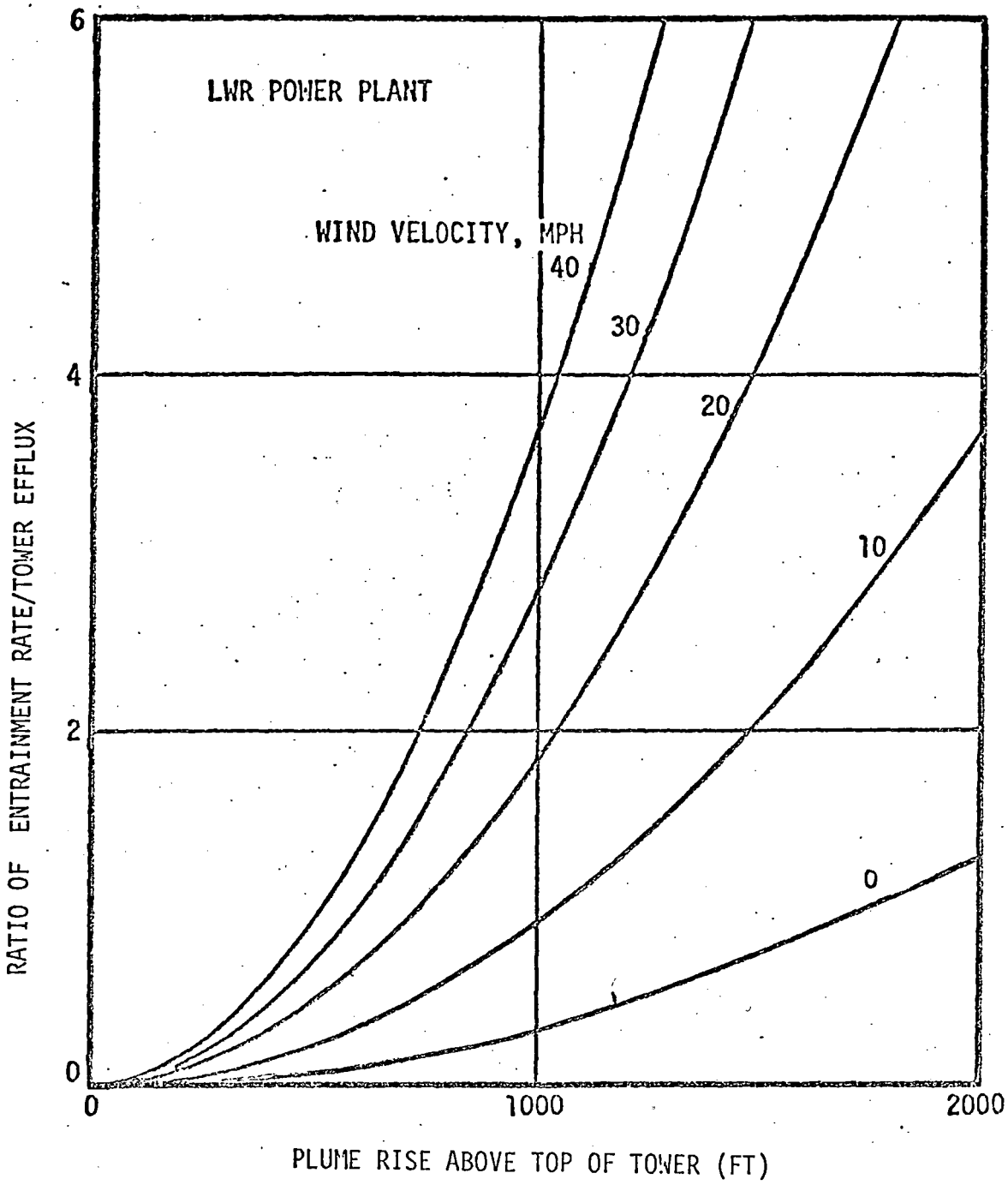


Figure 5-12. Entrainment ratio for a natural-draft cooling tower

5-4 Potential Environmental Impacts

The potential of a large dry cooling system to affect the environment in a region local to a power plant must be carefully considered. It has been suggested in the literature that large dry cooling towers might cause cloud and precipitation initiation, modification of wind and circulation pattern, and alteration of local microclimate through change in temperature or fog frequency. In this section attempts are made to summarize the available information pertinent to this subject. Even though the present knowledge does not permit us to fully determine all possible environmental impacts, it is still feasible to reach some tentative conclusions in some areas of interest to the public

5-4-1 Air Temperature

The plume temperature in the exit of a dry-type cooling tower is generally higher than the ambient air by 20 to 40^{o}F. The plume usually remains aloft and tends to dissipate over a wide horizontal area. Because of these dilutions, the plume would experience a gradual temperature drop in a large space. In general, there is no significant effect on the air temperature in the ground-level. If the plumes from dry-type cooling towers are such that they would penetrate the elevated inversion layer as previously described, the diffusion processes will take place at high altitudes and long distances. Under these conditions, the air temperature in the ground level is again not significantly affected.

Calculations have been done to confirm the above observation. Some of the results are presented in Fig. (5-13), Fig. (5-14), Fig. (5-15) and Fig. (5-16). The centerline temperature of the plume as a function of distance

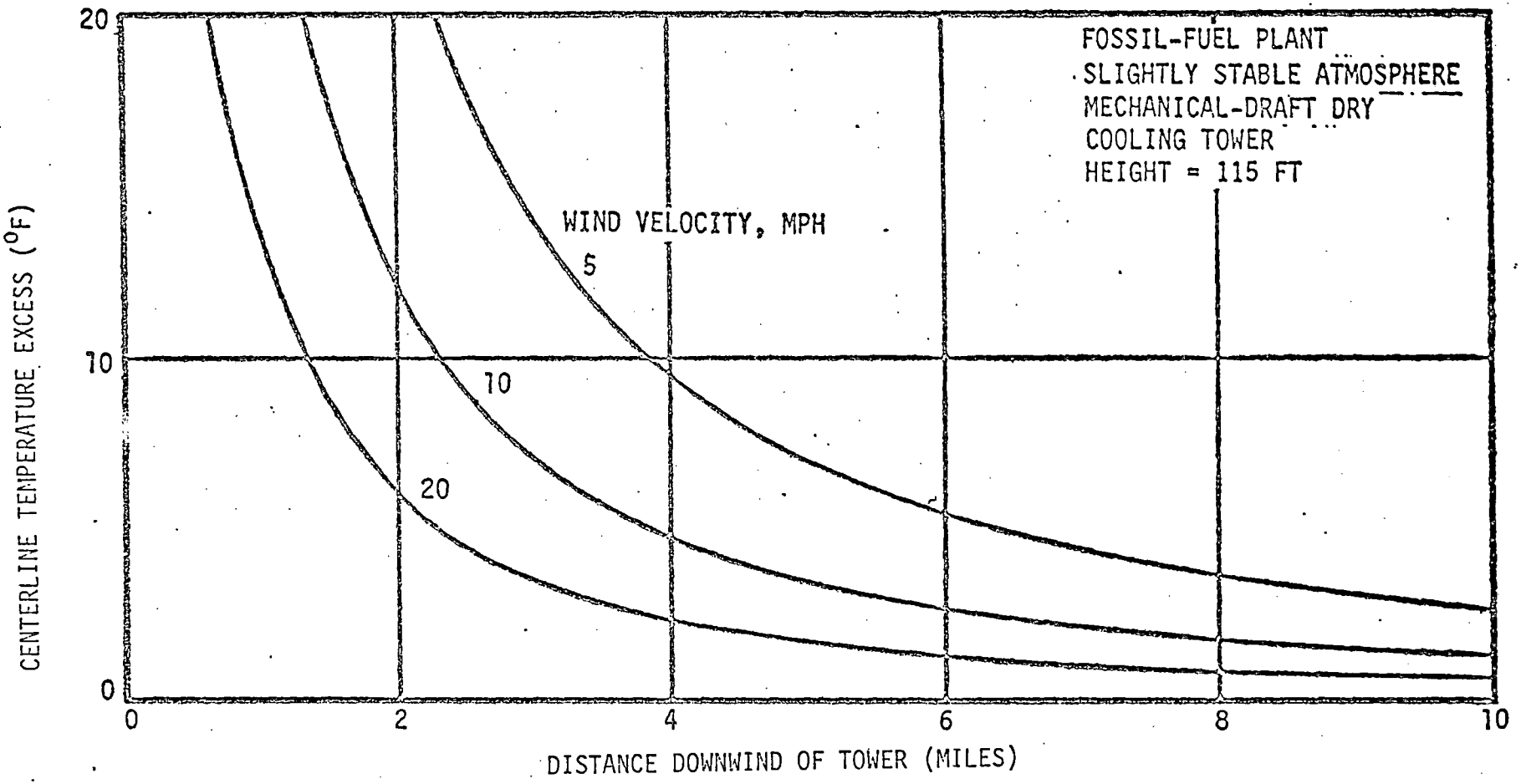


Figure 5-13. Centerline temperature excess over ambient temperature of a heated plume from a mechanical-draft tower

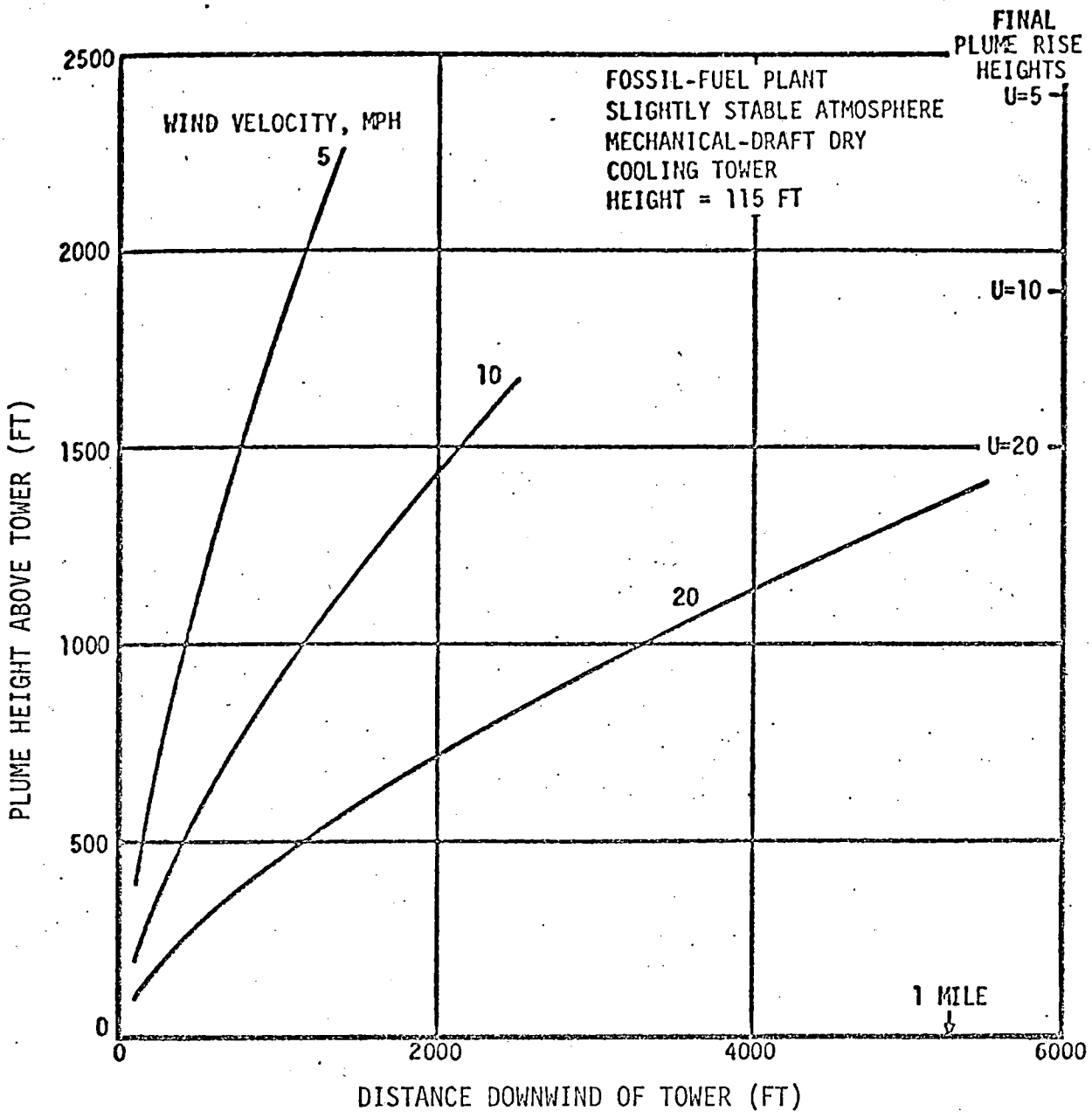


Figure 5-14. Initial rise path of a plume from a mechanical-draft tower

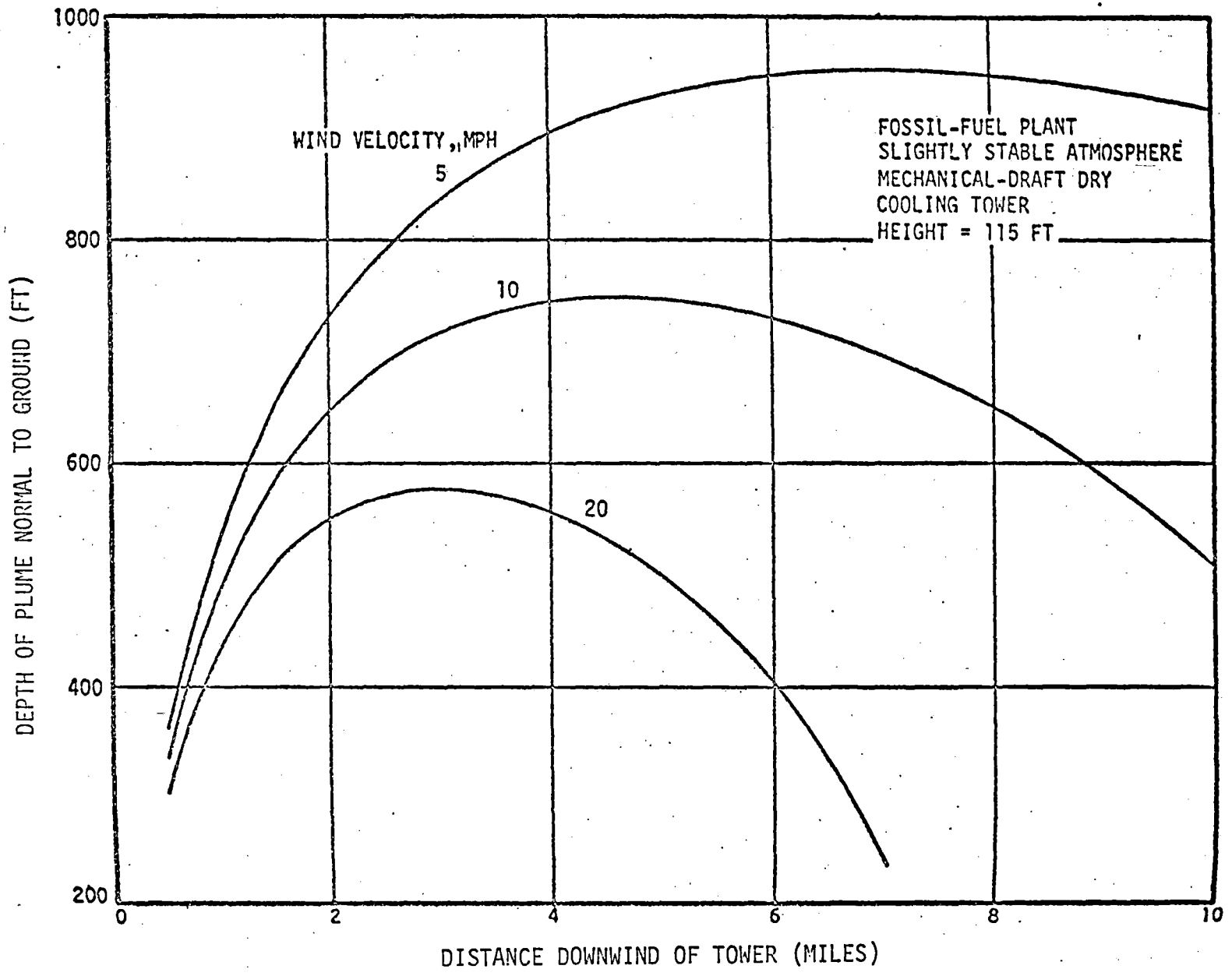


Figure 5-15. Extent of plume normal to the ground containing a temperature excess of $1^{\circ}F$ above the ambient

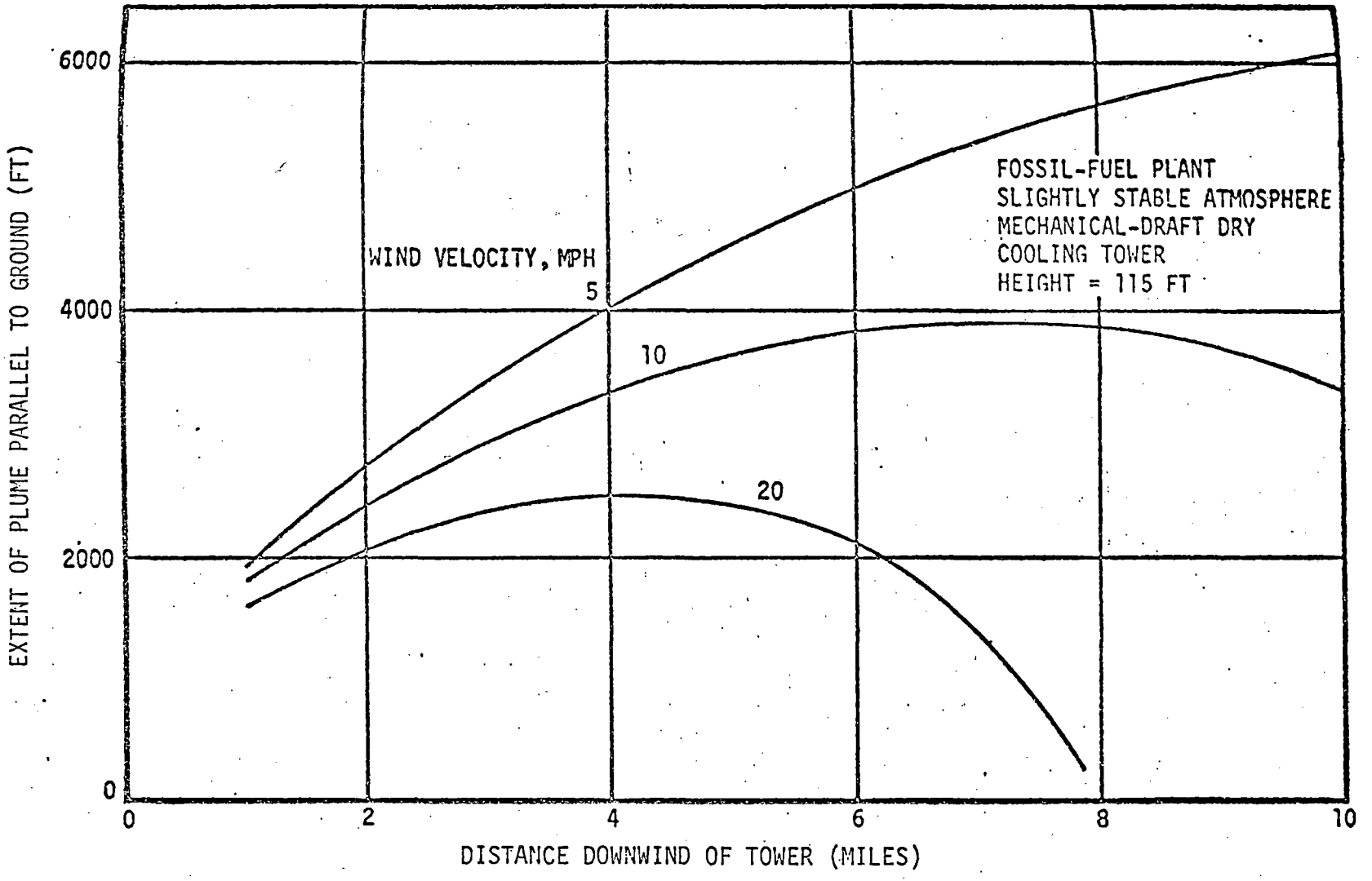


Figure 5-16. Extent of plume horizontal to the ground containing a temperature excess of 1 °F above the ambient

from the tower is in Fig. (5-13) Fig, (5-14) indicates the path of plume at various wind speeds. It is seen in these figures that the plume path is generally at high altitudes and the centerline temperature decreases very rapidly in the downwind direction. Fig. (5-15) and Fig. (5-16) show the plume spreadings both in horizontal and vertical directions. It is of interest to notice that the vertical plume spreading is much smaller than the horizontal plume spreading. Because of these, it is expected that the air temperature in the ground level is virtually unaffected. For example, at a wind velocity 10 mph, the plume center is at the altitude of 1900 ft at the downwind distance 5 miles. At the same location the plume spreading in a vertical direction is only about 750 ft. In other word the air temperature from the ground level to 1500 ft. level will probably remain unchanged.

It should be pointed out that the local air may still experience a temperature increase from a bent-over plume issuing from a relatively low mechanical-draft tower. However, based upon the above calculated results, the temperature increase is still a very minor thing.

5-4-2 Air Current

Dry cooling towers could affect local air movements in two ways. First, the influx of a large flow rate of air through the tower will result in detectable winds in the vicinity of the tower, and secondly, the plume might alter wind patterns in the surrounding region.

Fig. (5-17) and Fig. (5-18) show the induced wind near a large dry cooling system. At the distance 500 ft. from the tower, the maximum induced velocity is about 1 mph. Also shown in these figures are the

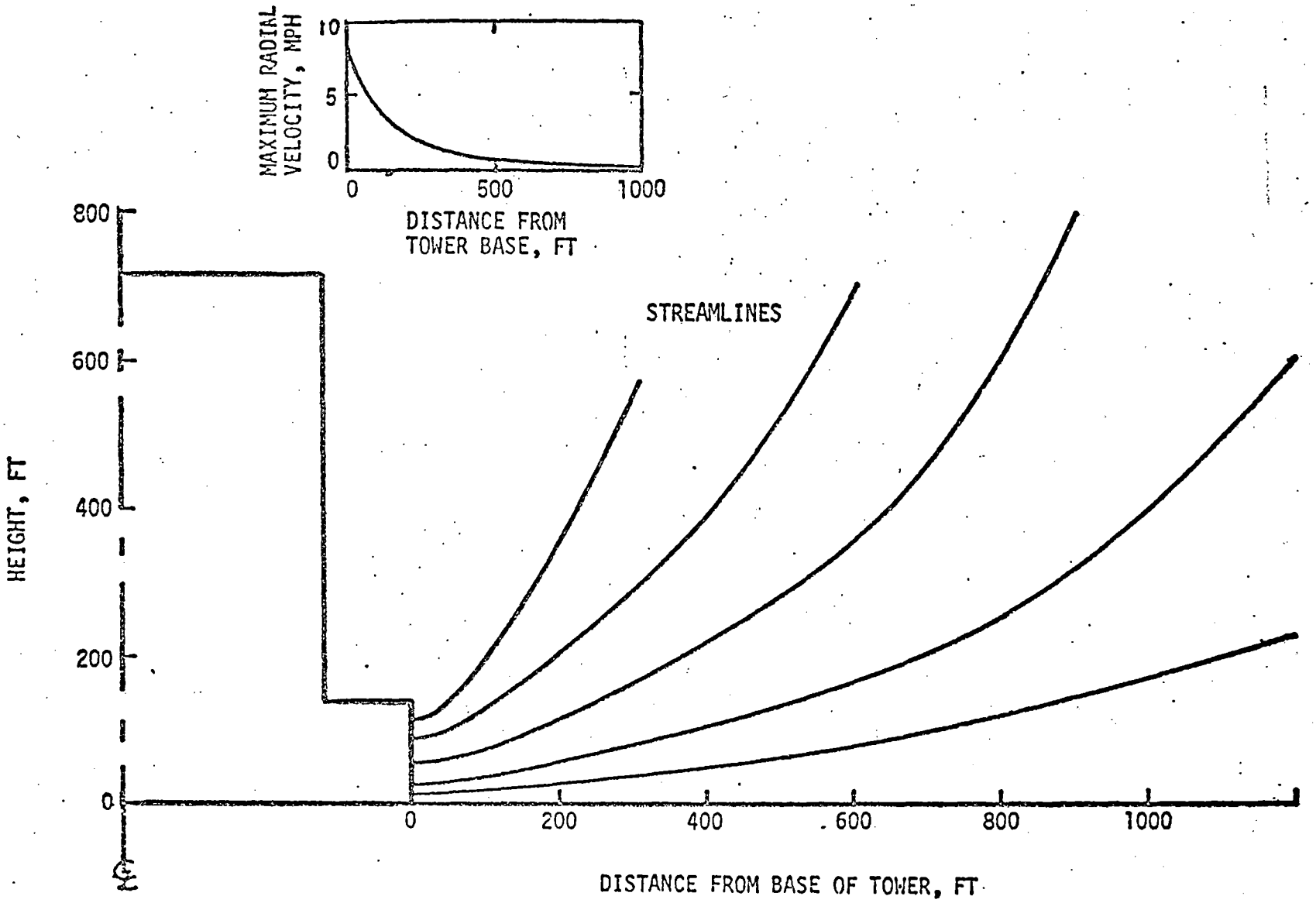


Figure 5-17. Induced wind near a large natural-draft dry cooling tower

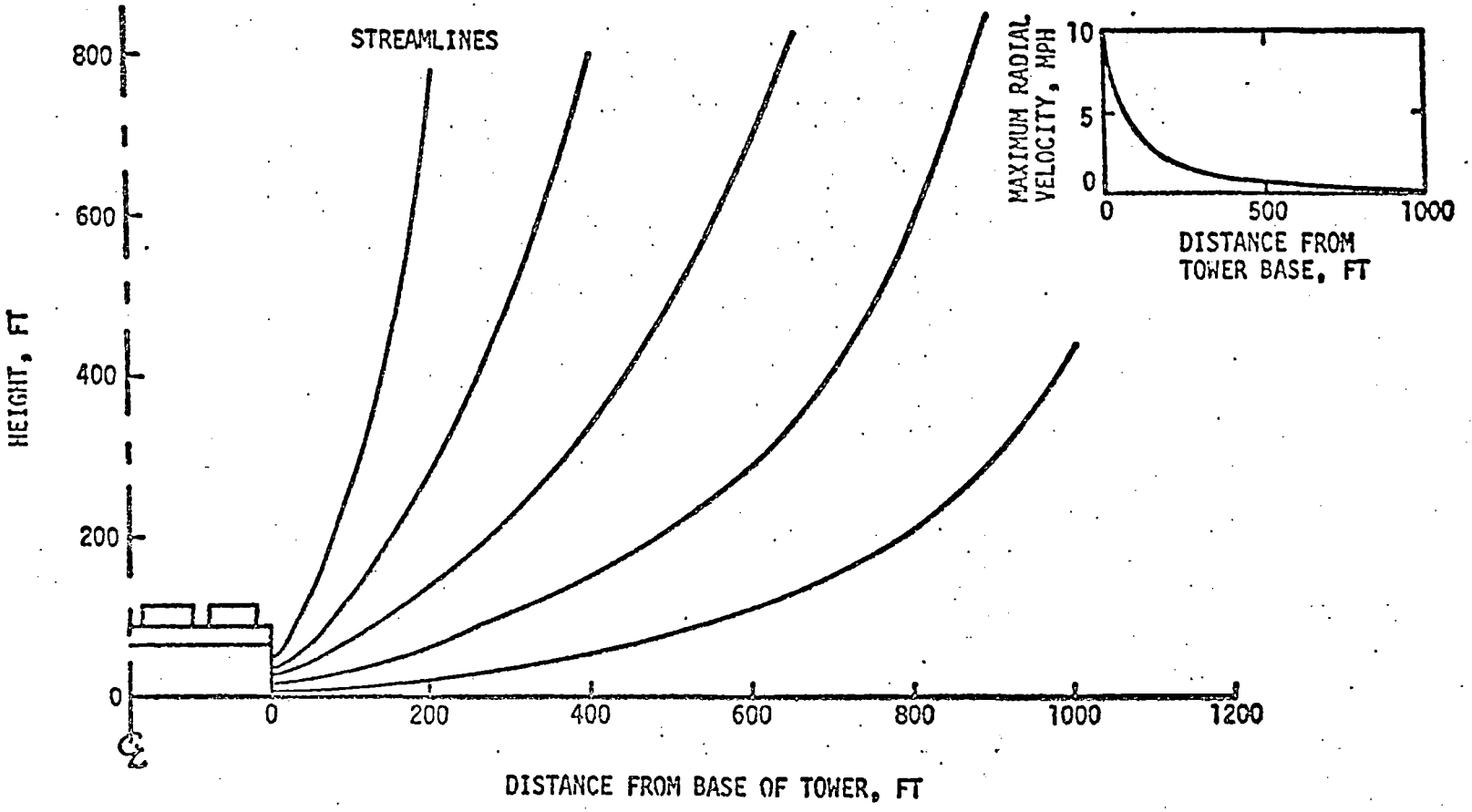


Figure 5-18. Induced wind near a large mechanical-draft dry cooling tower

streamlines of induced air movements.

The plume leaves a dry-type cooling system at a finite velocity. This exit velocity varies from one design to another. As indicated in previous section this exit velocities are about 26 fps for natural-draft cooling towers and about 17 fps for mechancial-draft towers. Because of interactions, the plume generally carries the surrounding air upward. It may reach up to 3000 ft. above the top of tower. These information can be found in plume rise calculations such as those in Fig. (5-6) and Fig. (5-7).

5-4-3 Fog

The exit plume from wet-type cooling towers will tend to induce local fog as presented in previous section. In the case of dry-type cooling towers, the plume is in lack of moisture and therefore, has a potential for dispersing fog. There are two mechanicsm by which fog dispersal would occur. First, the plume from a tower will be carried downwind where it can mix with the surrounding fog-laden air and so disperse the fog. Secondly, the air mass surrouding the base of the tower could be swept clear by the continued air ingress to the tower. In some sense the second mechanism does not disperse the local fog but prevents it by moving the air through the cooling towers.

Most fogs can be classified either as advection or radiation fogs. While there are some difference in terms of fog characteristics, these fogs generally occur in the lower levels. The advection fog usually appear in the atmosphere layer between the ground level and up to 700 ft. in altitude. The radiation fog which occurs in a small wind can be only found on the ground level and up to 150 ft. high.

With information of plume path presented previously, it is not difficult to realize that the plume from dry-type cooling towers may not have any practical chance to utilize its potentials of fog dispersal. For example, the radiation fog would not be affected directly by the plume which rises almost vertically. As to the advection fog dispersal, an examination of Fig. (5-14) indicate that the plume centerline will reach 1500 ft. high for a wind speed of 10 mph at a short downwind distance (around 2000 ft. away from the dry-type cooling tower) and therefore, has no practical chance to disperse the fogs.

5-4-4 Cloudiness and Precipitation

There are several important studies done in this area. At the present time, no definite conclusion is made and available. Czapski [16] reported that "several severe consequences of large thermal emissions can be foreseen:

1. Severe thunderstorm and even tornadoes can be caused in very unstable weather situations by dry and clean heat emission.

2. Rainfall will be increased downwind for a considerable distance by as much as 12% statistically.

3. Cumulus and even stratocumulus clouds will prevail most of time downwind from a large power plant."

It should be pointed out that the large thermal emissions considered by Czapski is in the range of 5000 mw. For a 1000 mw plant, it is doubtful that the precipitation pattern around the cooling tower will be altered. Also, it is doubtful that a cumulus cloud will ever be formed above a dry natural-draft cooling tower.

5-5 Noise Levels

Any consideration on the environmental impact of dry cooling towers will certainly lead to the identification of, at least, the following problems: 1) Effects of tower noise on the enviroment, 2) Effects of waste heat on the atmosphere. [38]

For a 1,000 mw power plant there will be around 32 60 ft-diameter fans needed for moving the air through the dry cooling towers. Naturally, these big fans will generate the noise (often defined as the unwanted sound). Combined with the noise generated by other parts in the cooling system, the tower noise would certainly be at a quite high level. During a visit to the Neil Simpson Power Station of the Black Hills Power and Light Company at Wyodak, Wyoming (related to other sponsored project), we found that the tower noise was, indeed, quite annoying even though the plant capacity is only 22 mw. Because of this potential concern, this section discusses the noise of cooling towers and methods for noise evaluation.

As with air pollution, water pollution, and thermal pollution, noise has increased insidiously since the beginning of the industrial revolution. Today, in most areas in which we live, noise is twice as loud as it was fifteen years ago. It will double again by 1980 unless we take some appropriate steps to muffle the unwanted sound. Noise is generally considered as a hazard to the health of people. It may contribute to hearing loss, hypertension, and high blood pressure. Sound may also affect the unborn child in the mother's womb. There has been considerable research work done on the effects of noise on man. In his book, Kryter [35] identified some 4,000 articles in this area and presents an excellent summary.

The Federal Government has been very active in noise control legislation. In May, 1969, the Federal Walsh-Healey Act was passed. This act specifies that no employee working in an industrial organization shall be exposed to more than 90 dBA over a daily eight-hour period, though higher levels are permitted for a short time. This act pertains to all industries having Federal contracts exceeding $10,000. On December 29, 1970, the Occupational Safety and Health Act of 1970 was signed into law by the President. This new law, effective April 28, 1971, will broaden the area covered by the Walsh-Healey Act. The standards promulgated under the Walsh-Healey Act are still effective under this 1970 act. More information about the Walsh-Healey Act is presented later.

In the noise control area, the local governments, state or city, have the authority to deal with their particular noise problems. In 1971, the North Dakota Forty-Second Legislative Assembly passed the House Bill (No. 1436). This bill authorizes the State Health Council to establish reasonable standards, rules, and regulations for noise control in the State of North Dakota and to report the progress in this respect to the Governor prior to the Legislative Session of 1973. It is believed that similar legislative programs are in progress in other states.

With public awareness of effects of noise and legislative programs in both Federal and State government levels, engineers have started to change their attitude toweard the machinery noise. In the past, suppression of factory and machinery noise were not normally undertaken until the noise level had been found dangerously high. Because of these "add-on" features, the amount of improvements were often limited by economic and functional considerations. Also, the techniques for control of noise were generally primitive and largely based on trial-and-error experimentations. This

approach, though successful in some cases, often resulted in a disappointment or failure. In contrast to these, the modern engineering approach is to predict and control noise level in the initial phase of design. With elaborate sound measuring devices engineers can now continuously monitor the noise level, identify, and eliminate any abnormal sound sources which may also represent symptoms of deterioration in the machinery.

5-5-1 Data on the Sound Levels in Dry Cooling Towers

The noise of dry cooling towers consists of noises from the following main sources:

1) Fan
2) Gear Box
3) Electrical Motor
4) Air Turbulence
5) Air Flow Over Fins
6) Water Flow Through the Valves and Other Restrictions
7) Pumps
8) Resonent Effect of Air-Cooled Plenum Chamber

To predict and control the sound level in a dry cooling tower, the sound spectrum should be obtained and analyzed for each component in the system. For the purpose of this investigation, howver, we are more interested in the sound level produced by the whole system rather than those produced by the components in the system. Recently, Shipes published the sound level data of the dry cooling towers manufactured by Hudson Production Corporation [38]. These data are reproduced here and presented in Fig. (5-19) and Fig. (5-20). The sound pressure level (SPL) has been unitized (dBA). It should be pointed out that each cooling unit which has

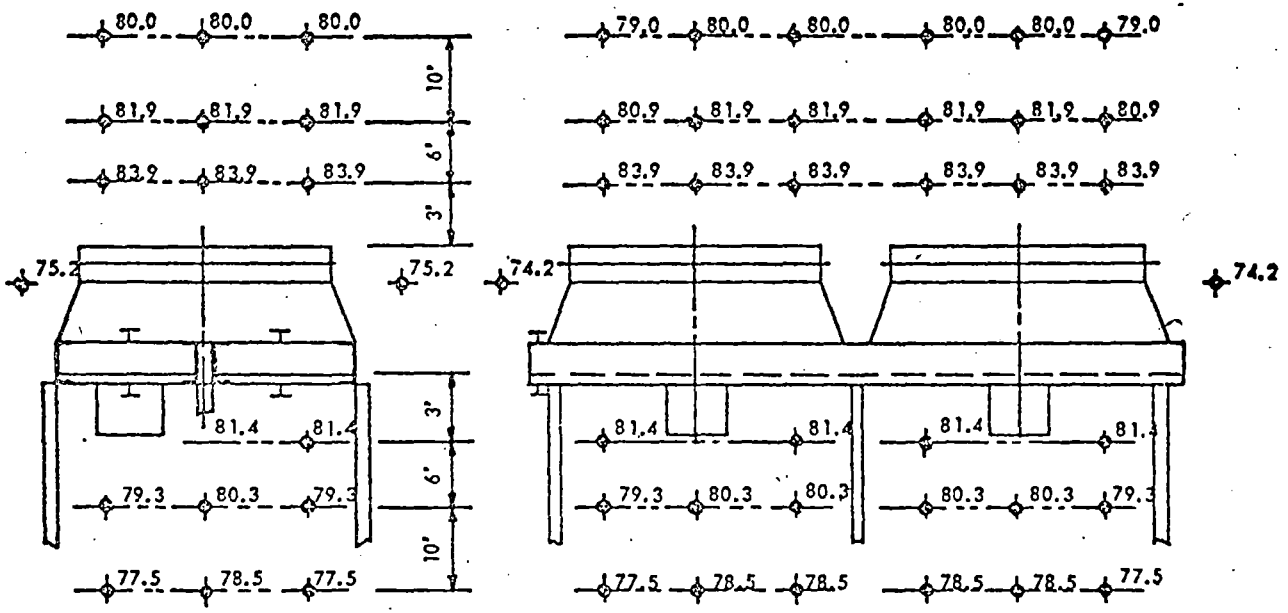


Figure 5-19. SPL readings near an induced mechanical-draft tower

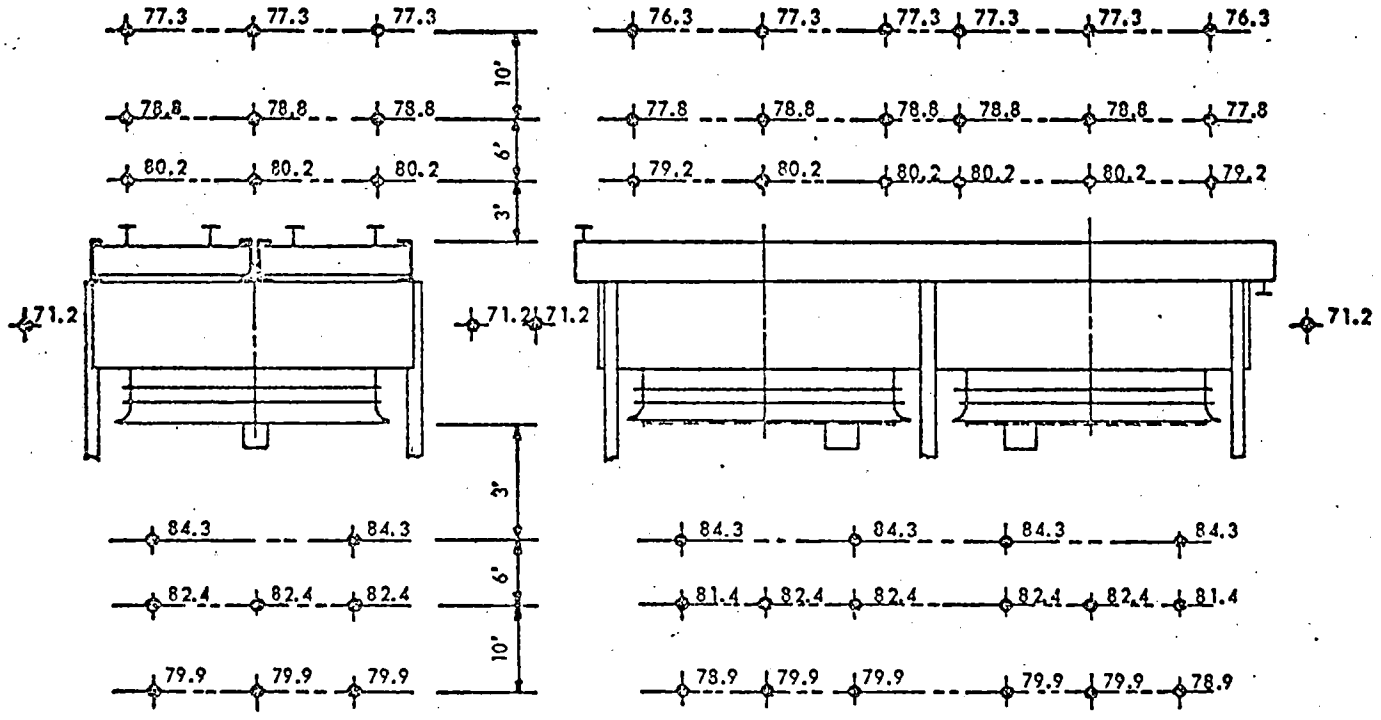


Figure 5-20. SPL readings near a forced mechanical-draft tower

two fans, was assumed operating under normal conditions and generating a sound power level of 100 dB at the time when the data were recorded. At the location, three feet away from the transition, the sound levels were found at the levels (75.2 dBA or 71.2 dBA). These are simply due to the shielding effects provided by the wall along the path of sound transmission.

Recently the fan noise (60 ft. diameter fan) has been measured and published by the same source (Hudson Product Corporation). The information is reproduced and presented in Table 5-4.

The sound pressure levels for a large mechanical-draft tower are presented in Fig. (5-21). Also shown in the figure is the comparison between the measured data and the standard noise conditions.

Baltimore Aircoil Company also published the sound level data on their products. (Note: the dry cooling towers manufactured by this company at this time are too small for power plant application.) They are shown in Fig. (5-22). Beside the differences in the data (which are certainly expected), the procedure and measuring method used by Baltimore Company are different from those used by Shipes. The lack of consistency is serious in industry and prevents us from meaningful comparison. It also causes confusion to engineers. At the 1972 Annual Meeting of the Cooling Tower Institute, a committee was established to determine a standard procedure and measuring method. In the next several sections of this report we will discuss different sound measurements for different purposes.

5-5-2 Sound Measurements Required by the Walsh-Healty Act

There are two sound measurements which are required and accepted by the Walsh-Healey Act. They are 1) by the standard sound level meter;

TABLE 5-4

ESTIMATED SOUND POWER LEVELS OF 60-FT FANS FOR MECHANICAL-DRAFT DRY COOLING TOWERS

Center Frequency (Hz)	PWL (dB re 10^{-12} W)	
	Single Fan	48 Fans
63	128	148
125	124	141
250	123	140
500	121	138
1000	120	137
2000	118	135
4000	115	132
8000	112	129

Notes:

1. Personal communication with K. V. Shipes of Hudson Products Corporation, Houston, Texas.
2. Fan tip speed = 12,000 ft/min.
3. Sound power level (PWL) deduced from sound pressure level data obtained at 0 to 500 ft from fan.

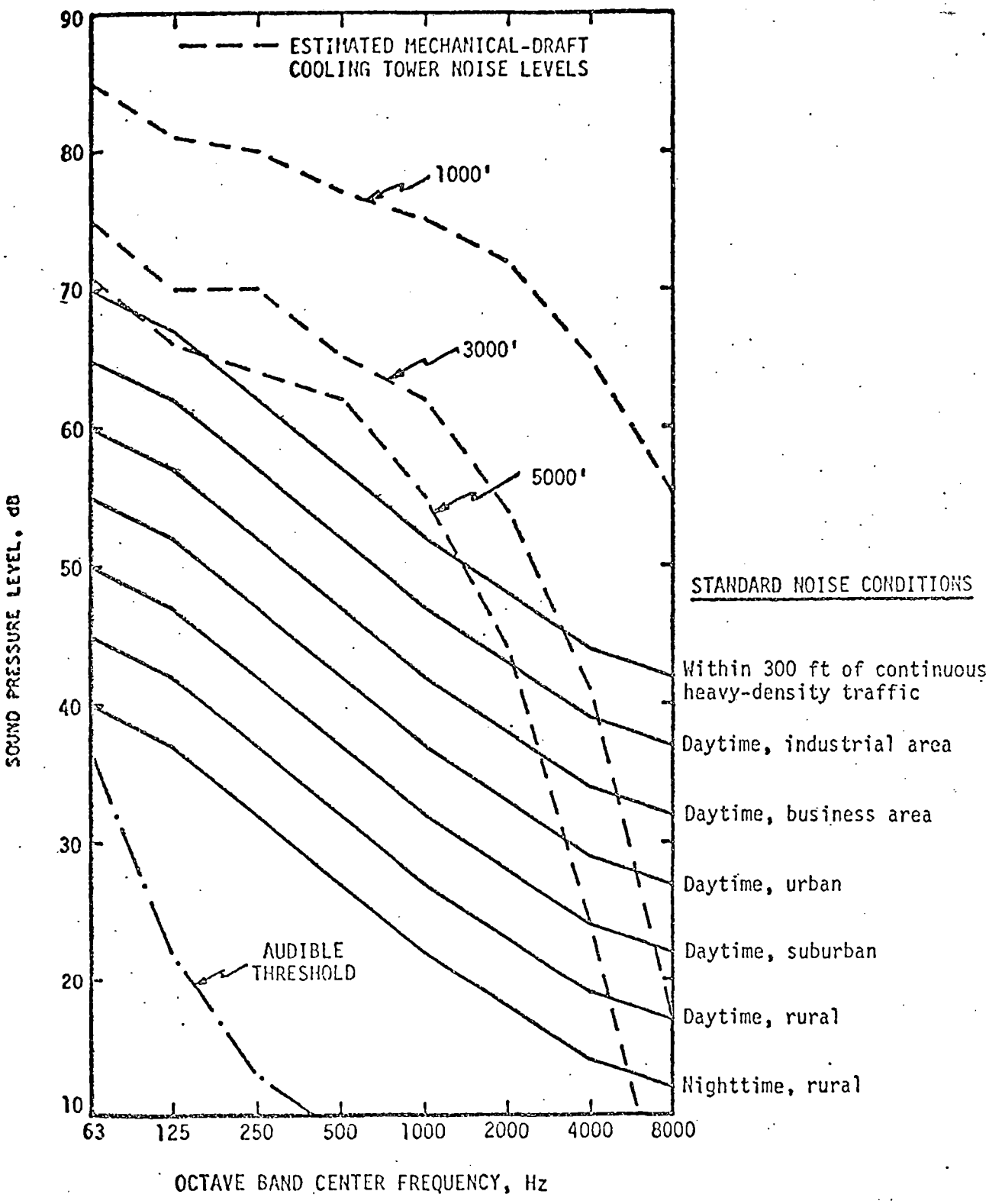


Figure 5-21. Sound pressure levels for a large mechanical-draft tower at various distance from the tower compared to standard noise conditions

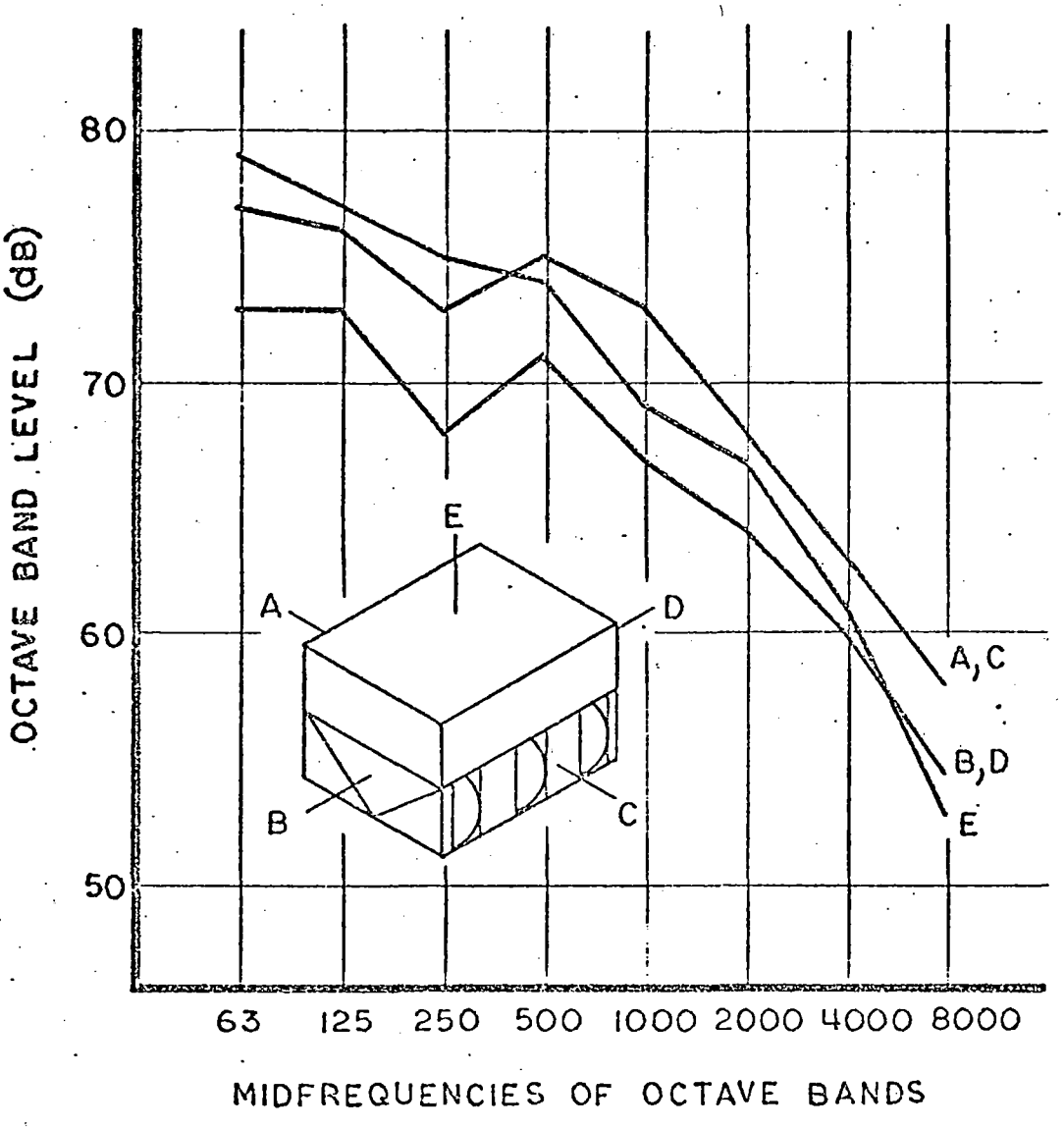


Figure 5-22. Noise spectrum of dry cooling towers

and 2) by the octave band sound analyser. When the standard sound level meter is used, it must be on the A scale at a slow response. The sound level thus obtained, shall not exceed those shown in Table 5-5. When the daily noise being measured is composed of two or more periods of noise exposure of different levels, the combined effect should be considered. This can be done by calculating the sum of the following fractions.

$$\text{Sum} = \sum_{i=1}^{n} \frac{C_i}{T_i} \qquad (5\text{-}8)$$

where C_i indicates the total time of exposure at a specified noise level and T_i indicates the total time of exposure permitted at that level. When the sum thus calculated exceeds unity, the mixed noise exposure is considered to exceed the limit value.

When the sound levels are measured by the octave band analyser, the octave sound pressure levels should be converted to the equivalent A-weighted sound level. Then, this equivalent A- weight sound level is compared with the limits shown in Table 5-5. This particular conversion can be easily done by plotting the octave bank sound pressure levels on a graph as shown in Fig. (5-23) and comparing the plotted curve with equivalent sound level contours.

Since the Walsh-Healey Act is concerned with the protection from the effects of noise exposure, the locations of measurements are naturally those where people work. If the sound levels exceed those permitted and the attempts to reduce sound levels fail, personal protection such as ear plugs or acoustical ear muffs should be provided. In this respect, it is interesting to know that the induced-draft dry cooling towers manufactured by Hudson would have a sound level less than 90 (dBA) and thus, would

TABLE 5-5

Permissible Noise Exposures Set by the Walsh-Healey Act

Duration Per Day, Hours	Sound Level dBA
8	90
6	92
4	95
3	97
2	100
1 1/2	102
1	105
1/2	110
1/4 or less	115

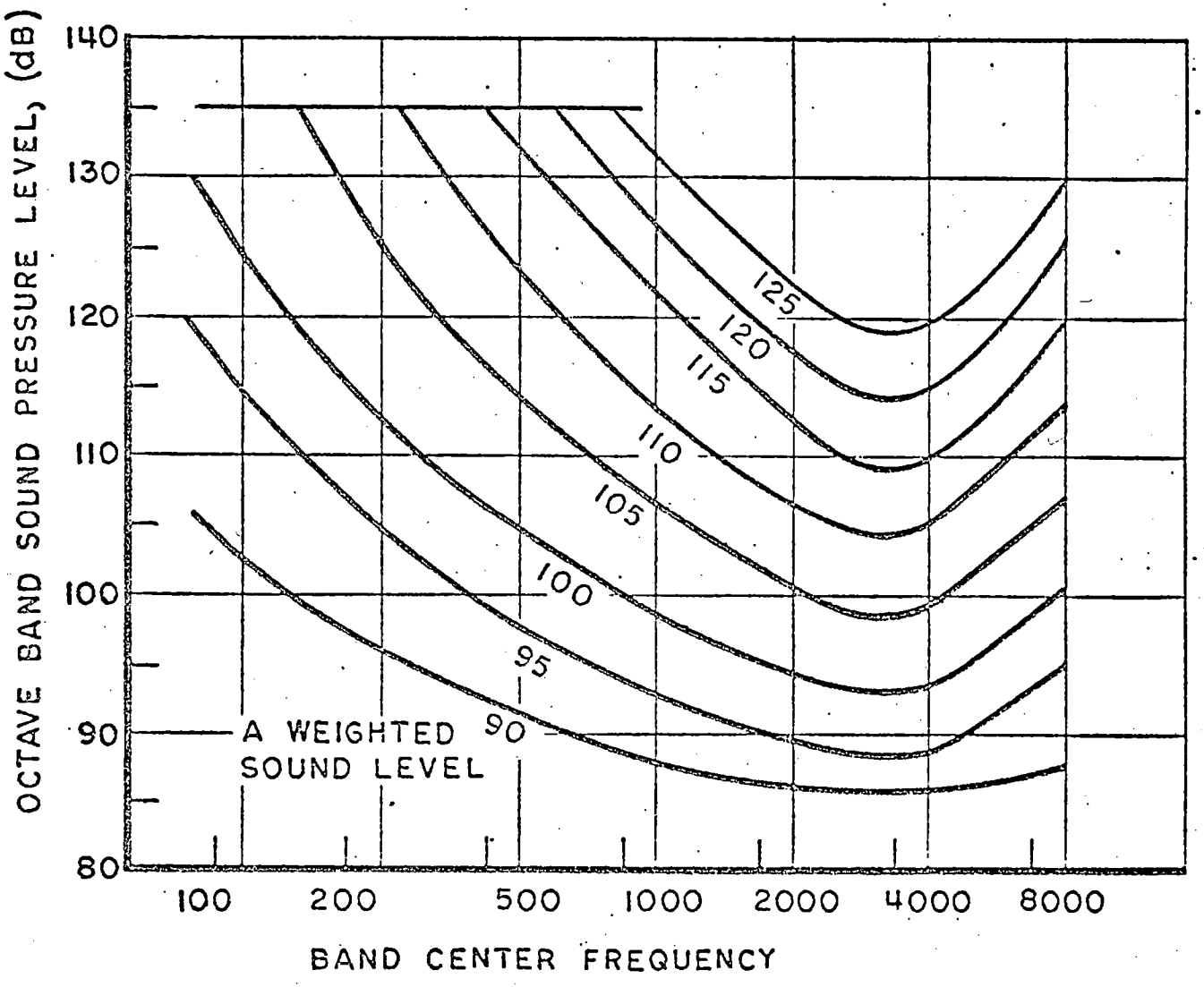


Figure 5-23. Equivalent sound level contours specified in the Walsh-Healey Act

satisfy the limit set by the Walsh-Healey Act.

It whould be realized that the sound level 90 (dBA) set by the Walsh-Healey Act is not the safest level. It is thought that at 90 dBA sound levels about 8% of the population will suffer some hearing impairment. At 85 (dBA) about 3% of the population will suffer some hearing impairment. In other words, the present permitted sound levels may be lowered if more consideration is given to the aspect of people's health.

5.5.3 Methods for Evaluation of Noise Environments

Sound level meters have the advantage of simplicity, but they often cannot give the answer we really want to know. It is because the broadband network in the meter cannot weigh a high-level low-frequency tone and a low-level high-frequency tone properly at the same time. For example, a sound meter would give a reading of 110 dB(A) for a 1 KHz tone and the same reading for a broadband signal which consists of an unsymmetrical square wave plus a sine wave. The actual loudness computed by the Zwicker method would be 128 Sones for the pur tone and 340 Sones for the broadband signal. The corresponding loudnesses levels are 110 phons and 124 phons respectively. These differences represent the "error" in the simple sound meter reading. To assess annoyance or nuisance value of noise, many methods have been proposed and used [38]. They are among many others:

1) Stevens' Method
2) Zwicker's Method
3) Kryter's Method
4) Noise Rating Method (NR Method)

In this study it has been found that the noise rating method is easily used in determining noises in and around power plants for acceptability with respect to annoyance. The locations of measurements would be those where the neighbors of power plants are. In this noise rating method, the noise spectrum is measured and compared against the noise rating curves shown in Figure (5-24). The noise rating number N of the noise is the number of the curve lying just above the spectrum. The interpretation and uses of noise rating numbers are presented in Table 5-6. To reduce the noise rating number, efforts should be concentrated on those frequency bands in which the octave band sound pressure levels are high.

Information about other methods for evaluation of noise environments can be found in References [38].

5.5.4 Sound Power Estimations

In the last two sections discussion has been centered on the sound measurements in noise environments to which people are daily exposed. These noise environments always consist of more than one noise source. In this section, however, the sound measurements to be described are classified as measurements of an individual noise source. These measurements become necessary especially when one cooling tower is compared with another tower. For this kind of comparison the sound power, rather than sound pressure, is used. Here, sound power means the total acoustic power radiated by a noise source (cooling tower). Evidently, this sound power cannot be directly measured as such. Instead, it is a quantity calculated from the measurement of sound pressure levels at various locations around the cooling tower. The measurement positions are shown in Figure (5-25). The equation used for

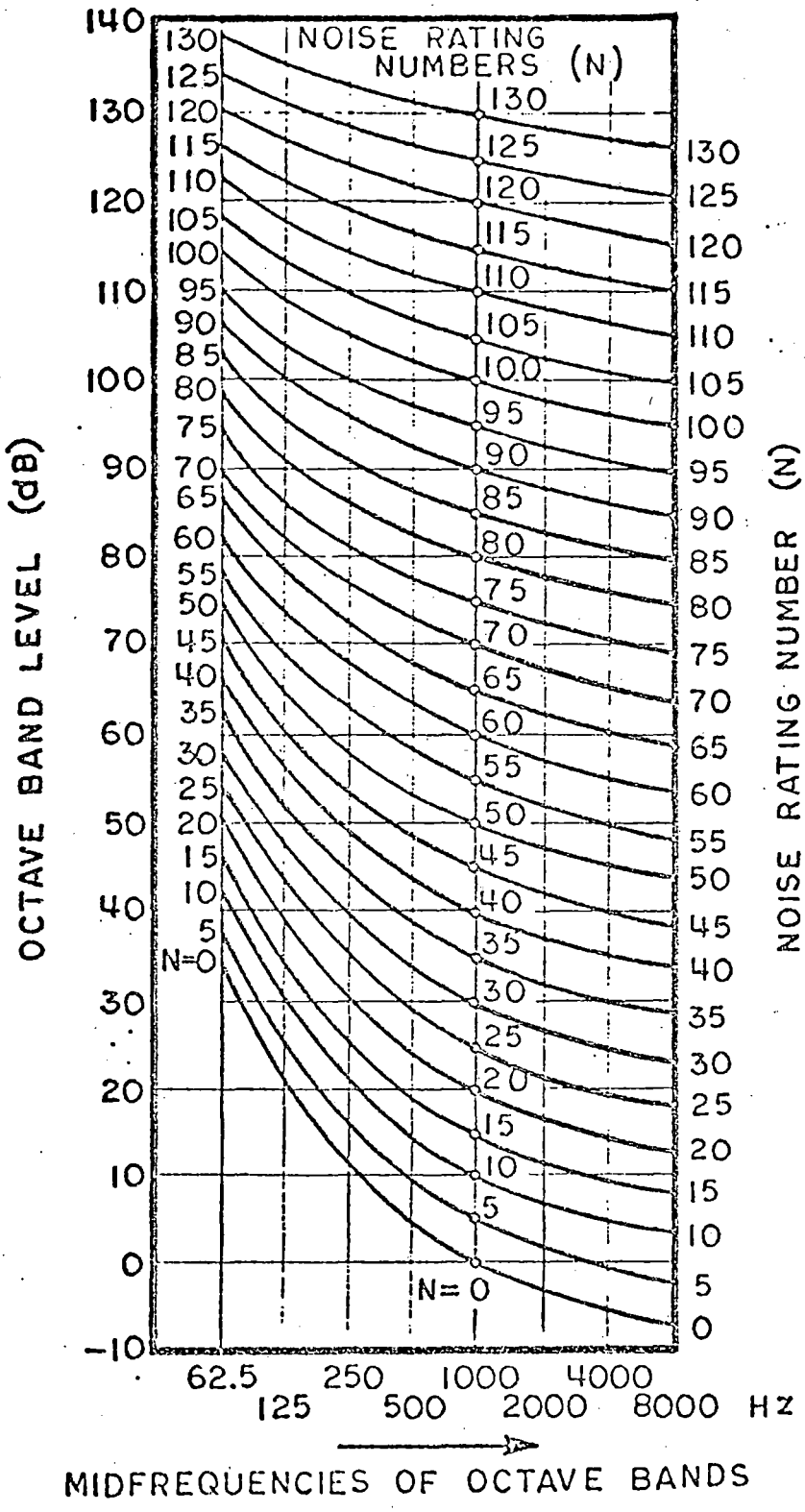


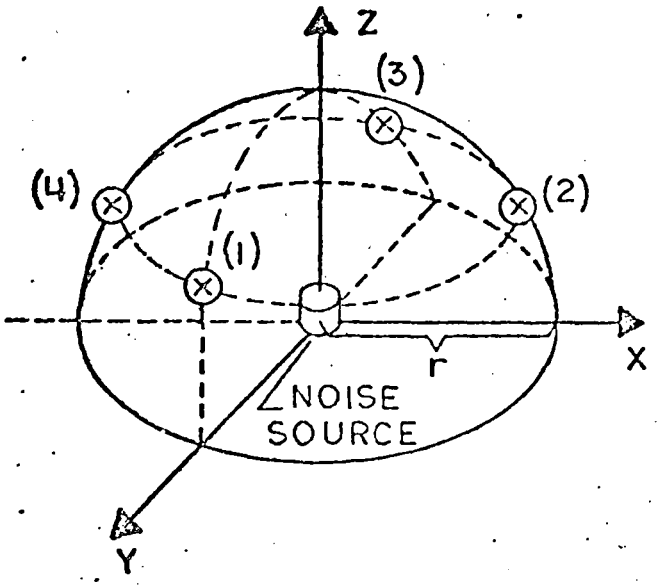
Figure 5-24. Noise rating curves

TABLE 5-6

Acceptability Criteria for Noise Rating Numbers

Location	Criterion In Terms of NR NO.
Broadcasting Studio	15
Concert Hall, Legitimate Theatre - 500 Seats	20
Classrooms, Music Room, TV Studio, Conference Room - 50 Seats	25
Sleeping Room (See Corrections Below)	25
Conference Room (20 Seats) or with Public Address System, Cinema, Hospital, Church, Courtroom, Library	30
Living Room (See Corrections Below)	30
Private Office	40
Restaurant	45
Gymnasium	50
Office	55
Workshop	65

Corrections for Dwellings	Change of NR NO.
a/. Pure Tone Easily Perceptible	-5
b/. Impulsive and/or Intermittent	-5
c/. Noise Only During Working Hours	+5
d/. Noise During 25% of the Time	+5
6%	+10
1.5%	+15
1.0%	+20
.1%	+25
0.02%	+30
e/. Very Quiet Suburban	-5
Suburban	0
Residential Urban	+5
Urban Near Some Industry	+10
Area of Heavy Industry	+15

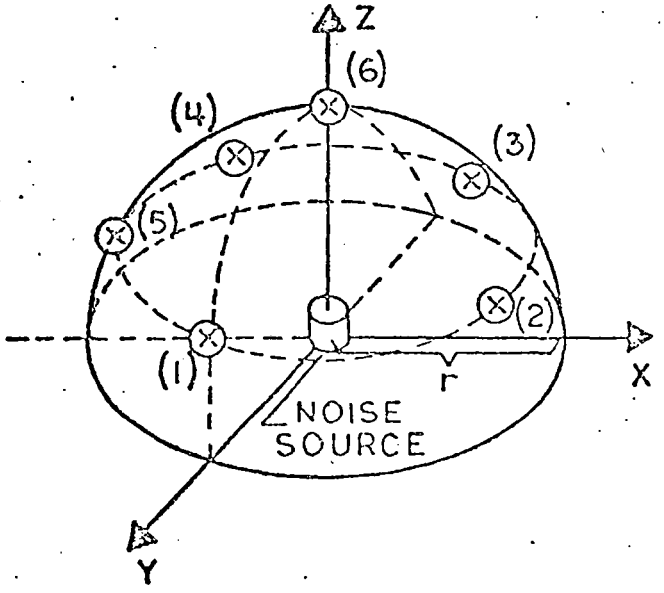


(a) 4 MEASURING POSITIONS

MICR. POS.	X	Y	Z
(1)	0	0.82r	0.58r
(2)	0.82r	0	0.58r
(3)	0	-0.82r	0.58r
(4)	-0.82r	0	0.58r

r = RADIUS OF HEMISPHERE

⊗ = MICROPHONE POSITIONS



(b) 6 MEASURING POSITIONS

MICR. POS.	X	Y	Z
(1)	0	0.89r	0.45r
(2)	0.85r	0.28r	0.45r
(3)	0.53r	-0.72r	0.45r
(4)	-0.53r	-0.72r	0.45r
(5)	-0.83r	0.28r	0.45r
(6)	0	0	r

Figure 5-25. The measuring positions over a hemisphere surrounding the noise source

estimation of the sound power level is:

$$PWL = 20 \log_{10} \frac{P_m}{P_o} + 10 \log_{10} \frac{2\pi r^2}{S_o} \qquad (5\text{-}9)$$

where P_m is the mean sound pressure

P_o is the reference sound pressure (0.0002 bar)

$2\pi r^2$ is the surface of the test hemisphere of radius r

S_o is the reference surface, $S_o = 1\ m^2$

It should be noticed that the sound power level in Eq. (5-9) is assumed relative to the reference power quantity 10^{-12} watt. The sound pressure ratio P_m/P_o is obtained with the relation:

$$\frac{P_m}{P_o} = \left[\frac{1}{n} \sum_{i=1}^{n} \left(\frac{P_i}{P_o}\right)^2 \right]^{1/2} \qquad (5\text{-}10)$$

where P_1, P_2 --- P_n are the sound pressures measured at the positions shown in Figure (5.25).

In many cases the tower (noise source) does not radiate the sound equally in all directions. To accurately describe the sound source, an additional quantity, the so-called directivity index, is needed. The equation for the directivity index in the i^{th} direction is:

$$D_i = 20 \log_{10} \left(\frac{P_i}{P_o}\right) - 20 \log_{10} \frac{P_m}{P_o} + 3\ dB \qquad (5\text{-}11)$$

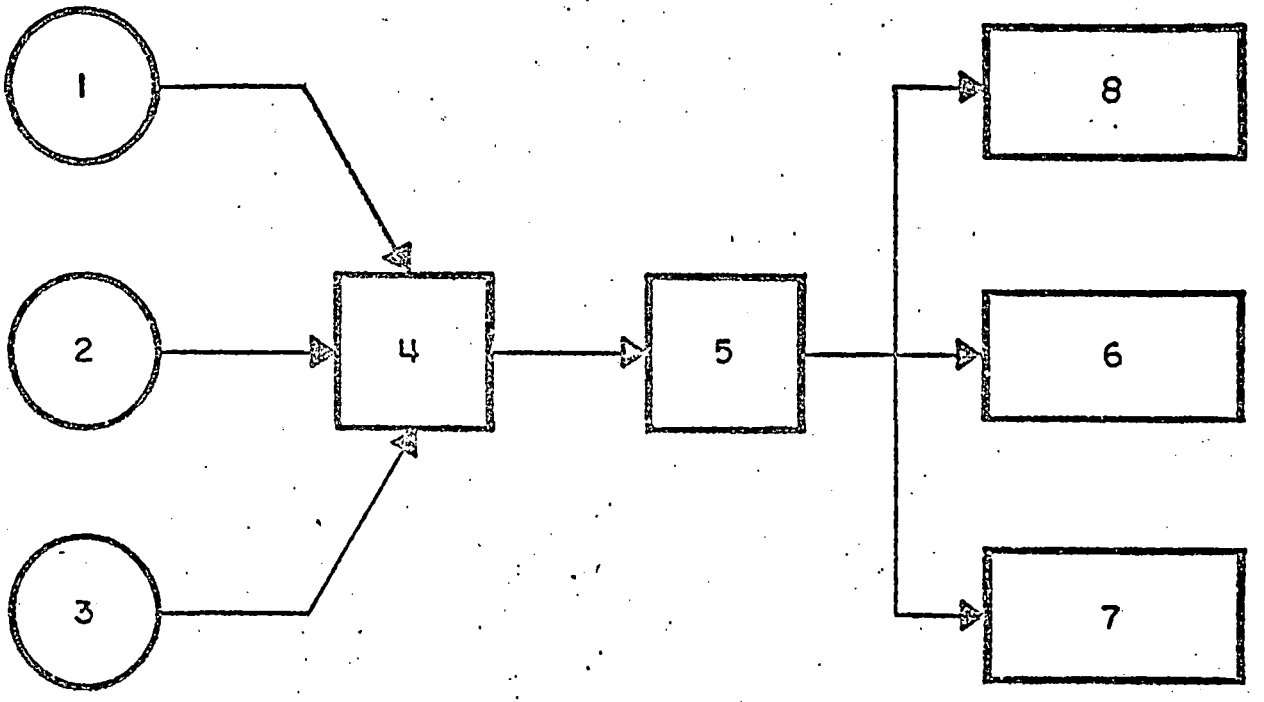
It should be pointed out that all equations presented in this section are based on the assumption that acoustically free field conditions would exist at the time of measurements. Also, the radius of the test hemisphere should be so chosen that it would be approximately equal to 2 to 3 times the largest dimension of the cooling tower.

5.5.5 Sound Measurement System for Power Plants

Besides the noise of cooling towers, there are many other noise sources in the steam-turbine plant. For a large power generating system the steam turbine and auxiliary equipment may represent other noise sources. To ensure an acceptable noise environment to the plant personnel and residents in the plant neighborhood (if any), a regular check on the sound level seems desirable and necessary. The sound level data thus obtained are believed useful in plant maintenance and noise control programs. Here is a brief description of a sound measuring system, needed for this purpose.

The measuring systems consists of eight channels of amplification, with band pass filters and recording equipment. The input may be either signals from microphones or from tape records. The display equipment of each channel consists of graphic level recorders, and statistical distribution analyzers in addition to conventional visual meter display. A diagram of a typical channel for sound measurements is shown in Figure (5.26)

The equipment may be employed in two different modes. In one mode, up to eight separate channels may be independently used, for example, in determining an overall sound level simultaneously at eight different microphone positions. In another mode, one microphone may feed eight channels in parallel permitting spectrum analysis in eight different frequency bands. In this system calibration signals are also provided for each channel.



1. MICROPHONE 2. TAPE RECORDER 3. CALIBRATION SIGNAL GENERATOR
4. AMPLIFIER 5. BAND PASS FILTER 6. VISUAL METER DISPLAY
7. GRAPHIC LEVEL RECORDER 8. STATISTICAL ANALYZER

Figure 5-26. Diagram of one typical channel for second measurement

All this measuring equipment can be easily mounted on a mid-sized vehicle, if necessary, forming a sound measurement mobile laboratory. This mobile laboratory would have the flexibility of use in almost any area where needed. Thus, it may be quite suitable for a power company which has plant facilities at several locations.

Reference

1. Amundson, T. E., "The Impact on Air Quality of Coal Consuming Facilities", M. S. Thesis, Department of Mechanical Engineering, North Dakota State University, Fargo, North Dakota, 1976.

2. Arndt, C. R. and R. E. Barry, "Simulation of Spray Canal Cooling for Power Plants - Performance and Environmental Effects", ASME paper 76-WA/HT-28.

3. Aynsley, E. and J. E. Carson, "Atmospheric Effects of Water Cooling Facilities", presented at the Cooling Tower Institute, 1973.

4. ASME Publication: "Cooling Tower Plume Modeling and Drift Measurement", published by the American Society of Mechanical Engineers, New York, New York, 1975.

5. ASME Publication: "Environmental Effects of Thermal Discharges", published by the American Society of Mechanical Engineers, 1970.

6. Baker, D. R., and H. A. Shryock, "A Comprehensive Approach to the Analysis of Cooling Tower Performance", Journal of Heat Transfer, (ASME), August, 1961.

7. Bhumralkar, C. M., "Computer Simulation of Atmospheric Effects of Waste Heat Rejected from Conceptual Large Power Packs", ASME paper 76-WA/HT-20, 1976.

8. Boyack, B. E. and D. W. Kearney, "Plume Behavior and Potential Environmental Effects of Large Dry Cooling Towers", published by Gulf General Atomic, February, 1973.

9. Brady, D. K., W. L. Graves, Jr., and J. C. Geyer, "Surface Heat Exchange at Power Plant Cooling Lakes", Research Project RP-49, Dept. of Geography and Environmental Engineering, The John Hopkins University, 1969.

10. Brennan, P. T., M. E. Smith, M. L. Kramer, and R. W. Reeves, "Behavior of Visible Plumes From Hyperbolic Cooling Towers", American Power Conference, Chicago, Illinois, 1976.

11. Carson, J. E., "Meteorological Effects of Evaporative Cooling Towers - Research Needs", ASME paper 74-WA/HT-sf, 1974.

12. Edinger, J. E. and J. C. Geyer, "Heat Exchange in the Environment", The John Hopkins University, 1965.

13. EPA Publication: "A Survey of Alternative Methods for Cooling Condenser Discharge Water", Part I, Water Pollution Control Research Series, 16130 DHS 07/69.

14. EPA Publication: "A Survey of Alternative Methods for Cooling Condenser Discharge Water", Part II, Water Pollution Control Research Series 16130 DHS 08/70.

15. EPA Publication: "A Method for Predicting the Performance of Natural Draft Cooling Tower", Water Pollution Control Research Series 16130 GKF 12/70.

16. EPA Publication: "Potential Environmental Modifications Produced by Large Evaporative Cooling Towers", Water Pollution Control Research, 16130 DNH 01/71.

17. EPA Publication: "Research on Dry-Type Cooling Towers for Thermal Electric Generation", Part I, Water Pollution Control Research Series 16130 EES 11/70.

18. EPA Publication: "Research on Dry-Type Cooling Towers for Thermal Electric Generation" Part II, Water Pollution Control Research Series 16130 EES 11/70.

19. Eynon, W. Y. and M. J. Margetls, "Natural Draft Cooling Towers - Design and Construction", presented at the Joint Power Generation Conference in 1972.

20. Gibbons, J. G., F. P. Pike and J. S. Neill, "A Cooling Pond Design Technique for Large Power Plants", presented at AIAA/ASME Thermophysics and Heat Transfer Conference, 1974.

21. Grimble, R. E. and A. Roffman, "Analytical Determination of Cooling Tower Drift Eliminators Efficiencies", presented at the Annual Meeting of the Cooling Tower Institute, 1973.

22. Grover, G. K. and R. K. Suri, "Transient Response of Parallel Flow Heat Exchangers", ASME paper 71-WA/HT-24, 1971.

23. Hanna, S. R. and J. Pell, "Cooling Tower Environment - 1974", published by Technical Information Center, Office of Public Affairs.

24. Harleman, D. R. F., editor, "Engineering Aspects of Heat Disposal from Power Generation", published by the Dept. of Civil Engineering, 1971.

25. Hauser, L. G., K. A. Oleson and R. J. Budenholzer, "An Advnaced Optimization Technique for Turbine, condenser, Cooling System Combination", presented at the American Power Conference, Chicago, 1971.

26. Holmes, R. E. and S. J. Basham, Jr., "A Dry Cooling System For Steam Power Plants", presented at the Intersociety Energy Conversion Engineering Conference, 1971.

27. Hosler, C. L., J. Pena and R. Pena, "Determination of Salt Deposition Rates from Drift from Evaporative Cooling Towers", presented in the annual meeting of Cooling Tower Institute, 1973.

28. Hoult, D. P., "Spray Pond Fog & Salt Deposition", Proceedings of 67th Annual Meeting Air Pollution Control Association, 1974, paper #74-10.

29. Howlett, L. D., "Large Scale Evaporative Cooling Of The Window Theory", ASME paper 76-JPGC-Pwr-1, 1976.

30. Huff, F. A. and J. L. Vogel, "Atmospheric Effects From Waste Heat Transfer Associated with Cooling Lakes", NSF G1-35841, 1973, Illinois State Survey, Urbana, Illinois.

31. Janson, H. J. and G. W. Underwood, "Design Application and Operating Experience with Cooling Towers on the AEP System", ASME paper 72-Pwr-3, 1972.

32. Juris, K., "The Operation of Mechanical-Draft Cooling Towers under Non-design Condition", presented at the annual meeting of the Cooling Tower Institute, 1972.

33. Knapp, C. E., "Thermal Discharges" Their Role in the Environment", presented in the annual meeting of the Cooling Tower Institute, 1971.

34. Kolflat, T. D., "Thermal discharge - An Engineering Problem", ASME paper 70-WA/PID-5.

35. Kryter, K. D., "The Effects of Noise on Man", Academic Press, 1970.

36. Ku, F. S. and R. L. Stubbins, "Effects of Mechanical Draft Cooling Towers on Heat Performance", ASME ppaer 72-WA/Pwr-15, 1973.

37. Li, K. W., "Cold-End Optimization Computer Programs", available in the Dept. of Mechanical Engineering, North Dakota State University, Fargo, North Dakota, 1976.

38. Li, K. W., "Dry Cooling Towers for Power Plants", published by the Northern States Power Company, Minneapolis, Minnesota, 1972.

39. Li, K. W., "Generalized Heat Balance Computer Program", available in the Dept. of Mechanical Engineering, North Dakota State University, Fargo, North Dakota, 1975.

40. Lin, Y. F., "Computer-Aided Design of Cooling Systems for Power Plants", MS Thesis, Dept. of Mechanical Engineering, North Dakota State University, Fargo, North Dakota, 1977.

41. Margetts, M. J. and F. M. Shofner, "Characterization of the Drift Emissions of a Natural Draft Cooling Tower and Examination of Sensitivity to Operational Parameter Variations", presented to Joint Power Generation Conference, 1973.

42. Murray and Trettel, "Interim Report on Meteorological Aspects of Operating the Man-Made Cooling Lake and Sprays at Dresden Nuclear Power Station", 1972, M & T Inc., Northfield, Illinois.

43. Neill, J. S. and J. H. Gibbons, "Sizing a Cooling Pond for a Power Plant", ASME paper 75-WA/HT-63, 1975.

44. Overcamp, T. J. and D. P. Hoult, "Precipitation From Cooling Towers in Cold Climates", presented in the annual meeting of Cooling Tower Institute, 1970.

45. Park, G. G. and P. S. Schmidt, "Heat Dissipation in a Power Plant Cooling Bay", ASME paper 72-WA/HT-61, 1972.

46. Parker, F. L. and P. A. Krenkel, "Physical & Engineering Aspects of Thermal Pollution", CRC Press Inc., 1970.

47. Porter, R. W., "Analytical Solution for Spray-Canal Heat and Mass Transfer", presented at the AIAA/ASME Thermophysics and Heat Transfer Conference, 1974.

48. Premchand, D., "The Comparative Study of Steam Power Plant cycle Arrangement of Digital Computers", M.S. Thesis, Dept. of Mechanical Engineering, North Dakota State University, Fargo, North Dakota, 1974.

49. Reddy, B. A., "Computer-Aided Power Plant Design", M.S. Thesis, Dept. of Mechanical Engineering, North Dakota State University, Fargo, North Dakota, 1975.

50. Roffman, A., "Drift Deposition Rates From Wet Cooling Systems" Proc. Symposium on Cooling Tower Environment, 1974.

51. Savery, C. W., "Counter-flow Evaporative Cooling Tower Performance Modeling and Verification", ASME paper 75-WA/HT-43, 1975.

52. Savery, C. W. and M. P. Hammill, Evaporative Cooling Tower Performance Predictions", ASME paper 72-WA/HT-62, 1972.

53. Sneck, H. J. and D. H. Brown, "Plume Rise From Large Thermal Sources Such As Cooling Tower", General Electric Report No. 72GEN025, 1972.

54. Soo, S. L., "Power Spray Cooling - Unit and System Performance", ASME paper 75-WA/Pwr-S.

55. Stephen, D. W. and W. J. Moroz, "Plume Rise From Wet Cooling Towers In Strong Winds", Engineering Research Bulletin B-107, The Pennsylvania State University, 1972.

56. Stewart, R. and R. T. Nelson, "Weather Modification Due To Spray Pond Cooling", ASME paper 75-HT-1, 1975.

57. The Marley Company, "Managing Waste Heat with the Water Cooling Tower", published by the Marley Co., 1970.

58. The Marley Company, "Drift Management in the Chalk Point Cooling Tower", published by the Marley Company, 1975.

59. The Marley Company, "Cooling Tower Fundamentals and Application Principles", published by the Marley Co., 1967.

60. The Marley Company, "Dry Towers and Wet-Dry Towers for the Indirect Power Plant Cycle", published by the Marley Company, 1974.

61. The Marley Company, "Drift Technology for Cooling Towers", published by the Marley Company, 1973.

62. West Associates Publication: "Dry Cooling Tower Study", published by Bechtel Corporation, April, 1971.

63. Wistrom, G. K. and J. C. Ovard, "Cooling Tower Drift, Its Measurement, Control and Environmental Effects", presented at the Cooling Tower Institute Annual Meeting, 1973.

64. Environmental Report for Sherburne County Generating Plant by Northern States Power Co. 1971.

65. Gulf General Atomic Publication Gulf - GA - A 12346 1973.

Property of
National FS Library
USDA Forest Service
240 W Prospect Rd
Fort Collins CO 80526